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HARVARD ENGINEERING SCHOOL

WAR-SHIPS

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A TEXT-BOOK

ON

THE CONSTRUCTION, PROTECTION, STABILITY TURNING, ETC., OF WAR VESSELS

BY

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WITH NUMEROUS DIAGRAMS

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PREFACE

THE present work has been prepared in response to suggestions made by a number of the senior naval officers taking the course in Naval Architecture at the Royal Naval College, Greenwich.

An attempt has been made to treat the subject from the naval officers' standpoint, and certain parts of the subject have been treated at some length with a view of meeting their special requirements. I am indebted to several officers for valuable suggestions as to the subject-matter likely to be of most use. The blank pages at the end have been inserted in order to provide space to note particulars and details peculiar to the ship on which an officer is serving, and to note changes of practice which may occur. The text is illustrated by a large number of carefully drawn diagrams. A number of questions have been prepared as an Appendix. These questions in many cases are designed to lead to inquiry and discussion, and cannot be directly answered from the text.

I am indebted to the Controller of the Navy, *Rear-Admiral W. H. May, M.V.O.*, for his permission to undertake the work, and to the Director of Naval Construction, *Philip Watts, Esq., F.R.S., LL.D.*, for his kindly interest in the undertaking. The book is, however, not an official publication.

Although prepared primarily for naval officers, yet it is believed that the work will prove a useful introduction to the subject of Naval Architecture for apprentices and students at the Royal Dockyards and elsewhere.

E. L. A.

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WAR-SHIPS

CHAPTER I.

THE STRENGTH OF SHIPS.

BEFORE considering in some detail the structural arrangements of various classes of ships in the Royal Navy, it will be of use to consider briefly the general nature of the strains to which a ship's structure is likely to be subject.

These strains may broadly be divided into two classes, viz.—

1. *Structural strains*, coming on a ship regarded as a complete structure, and

2. *Local strains*, which only affect a particular portion of a ship.

A ship may have great structural strength with small local strength; for example, a small torpedo-boat is strong enough to stand being lifted bodily out of the water on to a ship's deck, while the plating is very thin and easily damaged. A full-sized ship could not be thus lifted (supposing it to be feasible) without the probability of serious damage to the structure.

1. Structural Strains.—The most important of the structural strains to which ships are subject are those strains tending to bend them in a fore-and-aft direction. The ship may be regarded as a huge beam or girder. In order to bring out the various points in connection with this fore-and-aft bending and the proper arrangement of the material to make the ship strong enough, we shall first consider some properties of beams.

Beams.—If we take a plank, say 12 in. wide and 2 in. thick, and place it on supports 10 ft. apart, a ton weight placed at the middle would be as much as the plank could bear without

C.

breaking. If, now, we placed six such planks as in Fig. 1, the planks could stand 6 tons. If instead of simply resting on one another the planks are tightly clamped or bolted together, we should find that the beam thus made would stand far more than the six unconnected planks. If we carried the process a step further, we should have a log 12 in. wide and 12 in. deep of homogeneous material, and we should find that such a beam would stand 36 tons before it would be on the point of breaking, or six times the weight that could be carried by the six unconnected planks.

We notice that the upper layers of such a beam are being

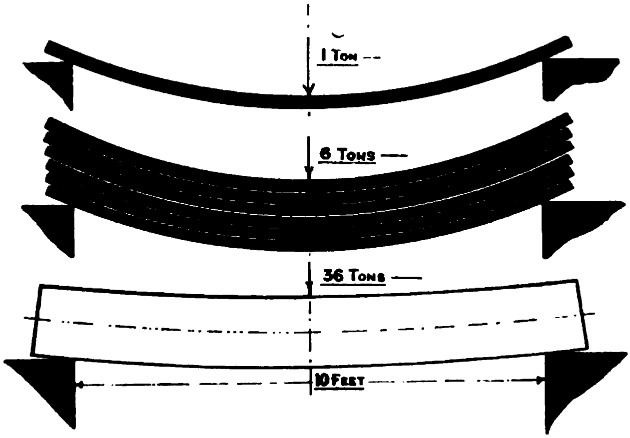

Fig. 1.

compressed and the lower layers are being stretched, and it is the resistance offered by the fibres of the beam, to this compression and this stretching, which enables the beam to withstand the bending. The middle layer of the beam is unaltered in length, and none of the upper layers are compressed to the extent that the top layer is, and none of the lower layers are stretched to the extent that the bottom layer is. So that if the timber of the beam is just strong enough to stand the tension at the bottom and the compression at the top without rupture, none of the layers between the top and the bottom are being used to their full strength. *Thus the portions of the beam furthest away from the centre of the section contribute most to the strength of the beam.*

The following will illustrate the same principle. Suppose it is desired to make an iron beam 12 ft. long, having a sectional area

of 16 square inches. How should this area be disposed to admit of the beam standing the greatest weight at the centre, without exceeding a stress of 10 tons per square inch on the material? If the beam is, as (*a*) Fig. 2, 8 in. wide 2 in. deep, it will stand a weight of about 1·5 tons; with a section, as (*b*), 4 in. wide 4 in. deep, it will stand about 3 tons; as (*c*), 2 in. wide 8 in. deep, about 6 tons, and as (*d*), 1 in. thick with flanges at top and bottom 5 in. wide, 10 tons. In the last case the bulk of the material composing the section is disposed most effectively in being away from the centre of the section, and although it has the same sectional area as (*a*), the beam can stand between six and seven times the weight.[1] A familiar instance of this principle is seen in the construction of many bridges. The upper and lower flanges are made exceedingly strong, and the web is often formed

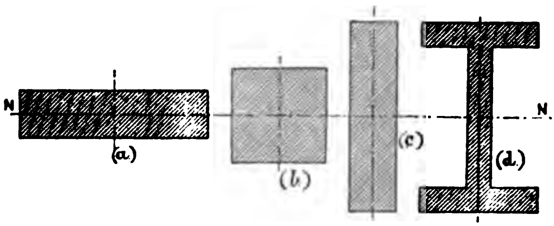

FIG. 2.

of lattice-work. (Charing Cross Railway Bridge in an instance of this construction.)

Longitudinal Strains.—In arranging the structure of a ship so that it shall efficiently resist the bending in a fore-and-aft direction, we arrange the material on the above principles. Special attention is paid to the sufficiency of the strength of the upper and lower parts of the ship, as the upper deck and the side plating adjacent, and the bottom plating, keel, longitudinals, etc., at the lower part.

The longer the ship is in proportion to her depth, the more necessary does it become to pay attention to these portions of the structure. Thus the upper-deck plating and side plating adjacent, and the structure at the keel, are made much stronger in a cruiser of 14,000 tons than in a battle-ship of 15,000 tons, because the proportion length ÷ depth is so much greater in the former case

[1] The thinning down of the web cannot, of course, be carried on indefinitely.

than in the latter. Fig. 3 shows in comparison the keel structure in these two cases.

In attempting to make these strains the subject of calculation we are confronted with the difficulty that it is not possible to

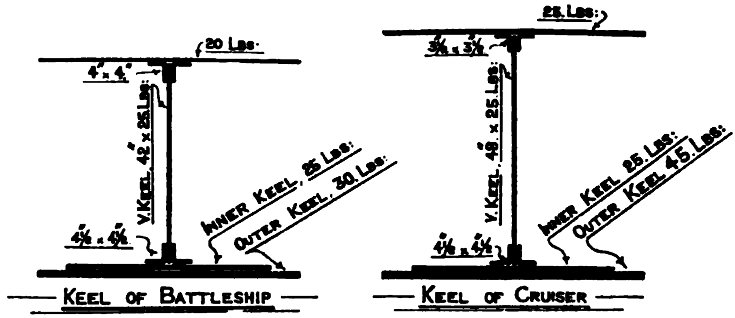

Fig. 3.

accurately determine the maximum strains the ship may reasonably be expected to withstand. An extreme case would be to assume that the ship is caught amidships on a rock, with the ends unsupported, or, again, if the ends only were supported. It would, however, be impossible to so construct a ship that the structure could stand anything so extreme as either of these conditions. The weight involved for the hull structure would be prohibitive.

For purposes of calculation, the following assumptions are made—(*a*) the ship is supposed to be momentarily poised on the crest of a wave of the same length as the ship, Fig. 4, and (*b*) astride

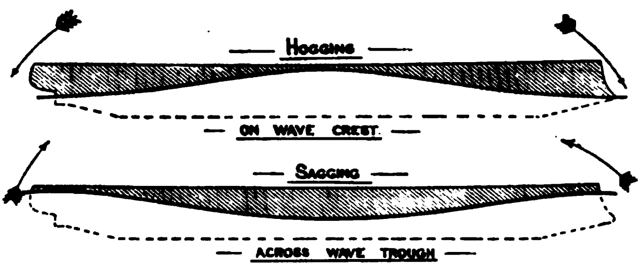

Fig. 4.

the trough of a wave of the same length, Fig. 4. Under the first assumption, the ends of the ship would *tend* to drop relative to the middle, and we should have the upper works *tending* to tear apart,

and the lower works to buckle up. Under the second, the reverse would be the case, viz. the keel and lower parts of the structure would *tend* to tear apart, and the deck and upper parts to buckle. The first is termed *hogging*, and the second *sagging*.

In a well-constructed ship these are only *tendencies*, and the material is able to withstand the strains thus brought to bear upon it. If, however, the ship is not strong enough, these tendencies will show themselves by the giving way of the structure at some point. In some destroyers, for instance, the compressive strain on the upper deck has caused the plating to buckle between the beams, and in these vessels it is most important to provide fore-and-aft stiffening to the deck, so that the plating will be able to stand up to the strain without deformation.

Even in still water there are strains on a ship, because the weight and the support of the buoyancy vary along the length. At the ends, for instance, there would be an excess of weight over buoyancy, because of the fineness of the ends, and at other portions of the length the reverse may be the case. These strains, however, are quite small in amount compared with those which might come on a ship in a seaway.

If the results of calculation on a certain ship on the above assumptions give a certain stress on the material, and the ship is found to show no signs of straining on service, it is safe to proceed with another ship, which by a similar process of calculation is found to be equally strong. The stresses on the material thus found are not regarded as absolute values. There are many conditions in the problem which cannot be taken into account, but the stresses thus calculated form a valuable means of comparison from one ship to another.

One feature in the construction of recent cruisers has been the adoption of special steel of high tensile strength. Instead of the tests specified for ordinary mild steel, viz. 26 to 30 tons per square inch, this special steel has to stand a tensile test of between 34 and 38 tons per square inch. This steel naturally is somewhat expensive, and is only used in special places, as *e.g.* for portions of the upper deck, and the upper and lower portions of the outer bottom plating. These are portions which, as seen above, are the most severely strained when the ship is hogging or sagging. Torpedo-boat destroyers of recent construction are built with decks and outer bottom plating of a steel of still higher tensile strength, viz. between 37 and 43 tons per square inch.

In deck-protected cruisers, like *Diadem* and previous ships, and present second and third class cruisers (see Figs. 21, 22, 24, 27), the heavy protective deck fitted in the region of the waterline, although valuable as a means of stiffening the ship, is not in a position to contribute anything like its fair share to the structural strength, because of its position near the neutral axis[1] of the section. In large cruisers since the *Diadem*, armour protection has been adopted at the side, and the main deck is made a protective deck as well as the middle deck (see Fig. 23). This is a much better distribution of the material as regards its usefulness in the structure of the ship, as well as for purposes of protection. The latest development in the design of large cruisers has been the adoption of a battery on the main deck instead of casemates, and the upper deck over this battery is made a thick deck. This deck is still better adapted to assist in the structural strength.

In large ships in the Royal Navy it is the established practice to build the structure mainly on the *longitudinal* system. That is, the longitudinal portions of the structure are made continuous, and the transverse portions are made in short pieces between, or *intercostal*. This system is carried out from the keel to the lower edge of armour or the protective deck, over the length of the double bottom (about two-thirds the length in a battle-ship). This system is admirably adapted to the formation of a double bottom. At the ends of the ship the vertical keel is still continuous, but the transverse framing is made the continuous part either side of the keel. The longitudinal strength is not so important at the ends, and is obtained by intercostal girders, the various platforms and bulkheads as well as the outer bottom plating. For smaller ships the transverse framing is made continuous and more closely spaced, and the fore-and-aft framing is mostly intercostal. The close spacing of the framing is necessary in order that the outside plating shall be well stiffened to hold it up to its work, as it is necessarily thin in a small ship. In these vessels a double bottom is not fitted. These features of construction will be more fully dealt with when we consider in detail the structural arrangements of various classes of ships.

If we compared two ships of the same size, one a war-ship and the other a merchant ship built to the rules of a registration society, we should find that the scantlings (or sizes of the steel used) of the former would be considerably less than those adopted

[1] The neutral axis is a horizontal line through the C.G. of the section.

in the latter case. There are several reasons which contribute to this. Merchant ships carry heavy weights of cargo in large holds, and the transverse framing especially has to be very massive to take the weight of the cargo. Again, merchant ships in some trades frequently ground when loading or discharging their cargo, and this again necessitates a good margin of strength as compared with war-ships, which are more carefully handled in this respect. Also, in war-ships a very extensive system of watertight subdivision is adopted; the large number of bulkheads and flats thus obtained assist very materially in the structural strength. Most merchant ships are built with the transverse framing for the most part continuous, with fore-and-aft girders intercostal. This system, although a very convenient method as regards economy in construction, is not so efficient from a structural point of view for large ships as the method adopted for war-ships. An important point in connection with the scantlings adopted in a ship is the subsequent maintenance. Inspection and care of the structure of war-ships is carried out in a most careful and thorough manner, and in this way the margin for deterioration may be made considerably less than in ships not so carefully looked after. The regulations for inspection of the structure of H.M. ships will be referred to later.

A very important principle in ship construction to be borne in mind is the necessity for avoiding any discontinuity in strength. Any part of the fore-and-aft structure which has to be ended, must be tapered down over several frame spaces, and a deep girder which has to be continued to the end of a vessel in a smaller form must be tapered down gradually to avoid any discontinuity of strength.

A superstructure like a boat deck is very high up, and if made a continuous portion of the structure is likely to be severely

Fig. 5.

strained, and it is not sufficiently strong to stand any severe strains. On this account the plating of this deck is deliberately cut through, and a sliding joint is made as Fig. 5 to save the

rupture of the deck that would otherwise occur. The side plating adjacent is cut by the gangway ports.

Transverse Strains.—The above remarks apply to the fore-and-aft structural strength; the transverse strength is also of importance. The rolling of the ship will tend to rack or distort the section, and this tendency is resisted by the transverse bulkheads, which are numerous in a war-ship because of the W.T. subdivision, and also by the connections of the beam arms to the frames, which are made specially strong for this purpose.

A ship when placed in dry dock, especially one with heavy weights of armour at the side, will have severe strains tending to tear the decks. Such a ship, however, is carefully shored as the water leaves so as to obtain plenty of support. It should be noted that the shores must be placed on transverse or longitudinal frames, or at bulkheads, in order to prevent the plating being injured.

2. Local Strains.—The above strains are termed "structural," because they come on the ship regarded as a complete structure, but there are a number of isolated strains that are brought to bear only on a portion of the vessel. Such strains are termed *local*.

Panting.—This term is used to denote the in-and-out working of the plating. It is especially found at the forward end of a ship which is subject to blows from the sea. If necessary, extra stiffening is added forward. In the large ships of the Navy panting is avoided because the forward end is made exceedingly strong to stiffen the bow and stem for ramming purposes, and also in some cases the armour protection is carried to the bow. The plating at the forward end is also well stiffened by the more closely spaced frames, and by the flats and platforms.

Severe local strains exist in the neighbourhood of heavy weights. Thus in way of an armoured barbette for the 12-in. guns, a tremendous weight is localized over a short portion of the length. Beneath such places the framing of the ship is made very strong, and bulkheads and pillars are arranged to distribute the strain throughout the structure.

Under the engines, where we have machinery in motion and not simply a dead weight, extra strength is necessary, and the framing, both longitudinal and transverse, is made very massive.

The thrust from propellers has to be transmitted to the structure of the ship by means of the thrust block. The fore-and-aft framing under the thrust is constructed very strong.

In way of gun mountings also, special strengthening is necessary, not merely to support the weight of the gun, etc., but also to withstand the recoil.

The security of masts at the decks where they are wedged has to be made of sufficient strength.

Bulkheads, both longitudinal and transverse, must be built of sufficient strength to withstand the strains due to the compartment on one side being full of water to several feet above the water-line. This would bring very severe strains on the bulkhead, and the safety or control of a ship might very conceivably depend upon one of the main bulkheads remaining intact under such circumstances.

Armour is found to be much more effective in resisting penetration when provided with a rigid support, and because of this the framing, etc., behind armour is of great strength. This framing is thus far stronger than would be necessary for the ship herself at such a part. It is locally very strong for the particular purpose of supporting the armour.

Another place of exceptional strength in a large ship is the stem and the structure adjacent. Here the strength is necessary for ramming purposes. This will be referred to at some length in Chapter VII.

CHAPTER II.

TESTS OF STEEL, ETC., SECTIONS, RIVETS, JOINTS, ETC.

AT the present time steel is the material almost universally employed in the building of ships. The steel employed is known as *mild steel*, which is a very pure form of iron, containing less than $\frac{1}{4}$ per cent. of carbon. Mild steel is quite a different material from the steel that is used for knives and tools. This tool steel is capable of taking a temper and hardening. Mild steel is produced of uniform quality and very reliable, and it is admirably adapted to stand the rather rough treatment in the shipyard, necessary in the formation of a ship's structure. A property of mild steel, which has frequently been found of immense service, is its capability of bending without fracture. There are many cases on record of steel ships having received severe injuries to the skin plating, but remaining quite intact. With wrought iron as formerly used, which is not so ductile, fracture is much more easily obtained. *High tensile steel* is being used in certain parts of cruisers and destroyers instead of mild steel. It is of greater strength, but is more expensive than mild steel, and it has to be carefully treated in working it into the ship to prevent reduction of the strength.

Tests of Material.—It is obviously of the greatest importance that the steel, etc., employed in the construction of the ships of the Royal Navy should be of first-class uniform quality, and in order to ensure this, a complete system of testing and inspection of material is arranged for at the works at which it is made. Only firms of good standing and having works of sufficient capacity are asked to tender, these firms being on the *Admiralty List*. The process of manufacture is carefully watched by the resident Admiralty Overseer; the following is an abstract of the tests carried out by the firm under his supervision.

Tests of Steel.—(*a*) *Tensile test of mild steel to determine the*

ultimate strength.—A strip is cut from the plate or bar selected by the Overseer, and it is planed to the shape shown in (*a*) Fig. 6, having a parallel width of 1½ in. over a length of 8 in. It is necessary that the edges should be planed and not sheared, as shearing makes the edges brittle. Such a test piece is placed in a testing machine, and should have a tensile strength of between 26 and 30 tons per square inch before fracture, and the stretching or elongation in the length of 8 in. should be 20 per cent., *i.e.* when the parts are put together, as in (*b*) Fig. 6, the original 8 in. should be 9⅗ in.

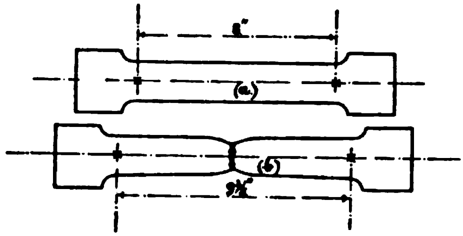

FIG. 6.

(*b*) *Bending test of mild steel to determine the ductility.*—This property of ductility is of importance for the reasons already stated. Strips 1½ in. wide, heated to a low cherry red and cooled in water of 80° Fahr., must stand bending double to a curve of which the inner diameter is three times the thickness. This test can be done cold if preferred by the Overseer.

For the *high tensile steel* used in cruisers, a tensile test of 34 to 38 tons per square inch is specified, with an elastic limit[1] of 20 tons. Similar results as to elongation and ductility are required as for mild steel. For the *high tensile steel* used in destroyers, a tensile test of between 37 and 43 tons per square inch is specified. The elongation and ductility required are rather under those given above for mild steel.

The Overseer selects one plate or bar for testing purposes out of every batch of fifty, or less than fifty.

In addition to the above specified tests, the Overseer makes an examination of the plates and bars to see that there is no lamination or flakiness, and that there are no objectionable hollows or other surface defects. For angle bars, etc., the Overseer can also test the bars at his discretion to ascertain the ductility under ordinary working conditions.

It is the Admiralty practice to specify plates by weight per

[1] Below the elastic limit the material is elastic, *i.e.* it will resume its original length if the strain be removed. Above the elastic limit this is not so, and "permanent set" takes place.

square foot, and angles, etc., by weight per lineal foot. Thus a steel plate ½ in. thick weighs 20·4 lbs. per square foot. A plate is ordered as 20 lbs. per square foot, and is thus really rather under ½ in. thick (0·49 in.). A steel angle bar specified as 3 in. × 3 in. of 7 lbs. per foot is rather under ⅜ in. There is a distinct advantage in ordering steel by weight in this way. We have an exact check on the thickness supplied, because by a simple measurement of the area or length we can tell what a plate or bar ought to weigh of the specified weight per square or lineal foot. Comparing this with the actual weight supplied, any excess or defect of thickness is at once apparent. Such variations of thickness are difficult to determine by actual measurement. Small excesses of thickness would amount in the aggregate to a considerable weight in a ship's structure. With plates 20 lbs. per square foot and over, and also bars, the manufacturer is allowed a latitude of 5 per cent. *under* the specified weight, and *no* latitude above. For plates under 20 lbs., a latitude of 5 per cent. *below and above* is allowed.

Treatment of Mild Steel, etc.—There are certain precautions necessary in the treatment of mild steel. Plates and bars should be heated as little as possible; when heated, no work must be done when the temperature has fallen to a *blue heat* (600–400° Fahr.). At this temperature the steel is very brittle, and the plate or bar must be reheated to complete the work.

When holes are punched in steel, it is found that the strength per square inch of the material left is considerably less than the original strength. The forcible entry of the punch makes the steel round the hole brittle. It is found, however, that the process of riveting partially restores the strength, the influence of the hot rivet and the subsequent hammering must alter the structure of the steel round the hole. The strength is restored if the plate is *annealed* (heated to redness and allowed to cool slowly under a heap of ashes). For the above reasons it is laid down that butt-straps of important parts of the inner and outer bottom plating, stringers, etc., should either (*a*) have their holes drilled, or (*b*) be annealed after the holes are punched. The former is not adopted in ordinary work on account of the greater cost.

The high tensile steel mentioned above as used in cruisers and destroyers has all the holes drilled. Punching is found to cause serious depreciation of the strength.

There are many holes in a ship's structure which have to be *countersunk*, *i.e.* the hole is made conical to take the point of the

rivet (see D, Fig. 9). In this case the injured material round the punched hole is removed by the countersinking drill, and the strength of the remaining material is fully equal to that of the original.

Pickling.—It is found during the process of manufacturing steel plates that there is formed on the surface a black oxide or scale, called *mill scale*, which clings very tenaciously to the surface. If, however, the plate is damaged, the scale will peel off and take with it the paint, leaving the bare steel underneath liable to corrosion. This scale also is electro-negative[1] to the steel, so that if moisture gets in under the scale, corrosion of the steel will go on very rapidly. It is, therefore, most important to remove all traces of the mill scale before the steel is painted. In Admiralty work it is the practice to immerse the plates on edge for a few hours in a bath containing dilute hydrochloric acid (1 part acid to 19 parts water). This loosens the scale, and the plates when removed are brushed with wire brushes and washed with a hose to remove the scale. The portions of the structure thus *pickled* are those liable to come into contact with sea-water or water in the bilges. These are the plates of the outer and inner bottoms, lower plates of bulkheads, plates of fresh-water tanks, and the lower plates of frames.

Rivet Steel and Rivets, Tests of.—The efficiency of the structure of a ship must ultimately depend on the quality of the rivets used to connect the various portions of the hull together. The following is a summary of the tests made to ensure having rivets of first-class quality:—

The steel bars from which the rivets are made must stand a tensile test of between 26 and 30 tons per square inch, with an elongation of not less than 25 per cent. in 8 in. For the bending test, the bar, either cold or heated as for mild steel, must stand bending double over a block having the same diameter as the bar. These tests are seen to be rather more severe than those for mild steel.

The rivets themselves, when made, must cool slowly, and samples taken at random must stand the following tests:—

(i.) A cold test as (*a*), Fig. 7.
(ii.) A hot test as (*b*), Fig. 7.
(iii.) A hot test as (*c*), Fig. 7, the head being flattened out to two and a half times the diameter of the rivet without cracking.
(iv.) The shank is nicked and bent over to show the structure of the steel.

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[1] *I.e.* If an electric current is set up between the scale and the steel, the steel will corrode, like the zinc does in the zinc-copper cell.

Rivets for use with high tensile steel are of special quality, so as to utilize the greater strength of the steel. These rivets are made with three projections on the head, as P, Fig. 9, to distinguish them from ordinary rivets.

Steel Castings, Tests of.—There are many parts of a ship's structure which formerly were made of iron forgings; these can now be more conveniently made of cast steel, and this material is always used in steel ships for the stem, sternpost, rudder-frames, shaft brackets, etc. The following are the tests laid down to ensure satisfactory castings:—

On each casting three projections are cast, to provide, when cut off and planed, test pieces 1 in. square. One of these must stand a tensile test of 26 tons per square inch, with an elongation of 10 per cent. in 8 in. One must stand bending to an angle of 45° over an edge having a radius of 1 in. The other

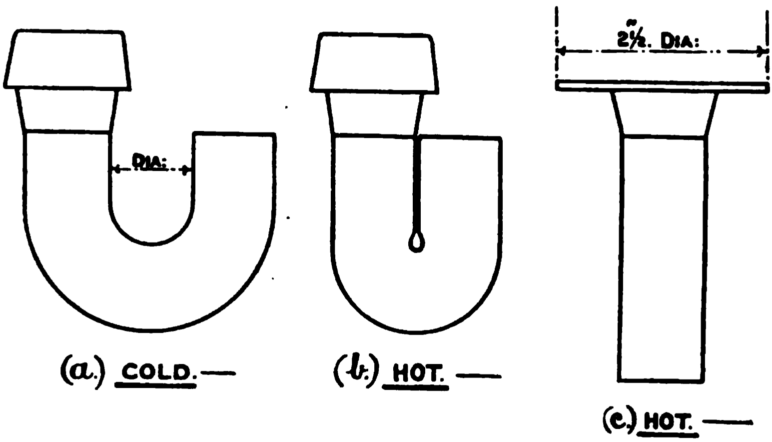


FIG. 7.—Rivet tests.

test piece is available if the results of either of the others are disputed. It will be noticed that these tests are less severe than for mild steel.

Although the material of the casting may satisfactorily stand the above tests, it is most necessary to find out, if possible, whether any blow holes exist in the body of the casting. In order to thoroughly shake up the casting, it is either (*a*) raised to an angle of about 60°, or (*b*) lifted bodily to a height of about 12 ft. and dropped on hard ground. The latter would be adopted for small castings. The casting also is suspended in chains and hammered all over with a heavy sledge to find out if it rings true; the surface is also carefully examined for defects.

Phosphor Bronze Castings, Tests of.—When a vessel is sheathed with wood and copper, it is not possible to use cast

steel for the stem, sternpost, etc., because of the galvanic action that takes place between copper and steel, if metallic connection is established between them. For such ships, therefore, the stem, sternpost, etc., are made of the copper alloy, *phosphor bronze*. This consists of 90 per cent. copper with 10 per cent. of phosphor tin (containing 5 per cent. of phosphorus). This material should stand a tensile test of 17 tons per square inch, with an elongation of 15 per cent. in 6 in. This is seen to be considerably lower than for cast steel. A chemical analysis of some drillings is made to see how near the material is to the composition specified.

Sections of Steel.—One great advantage attending the use of

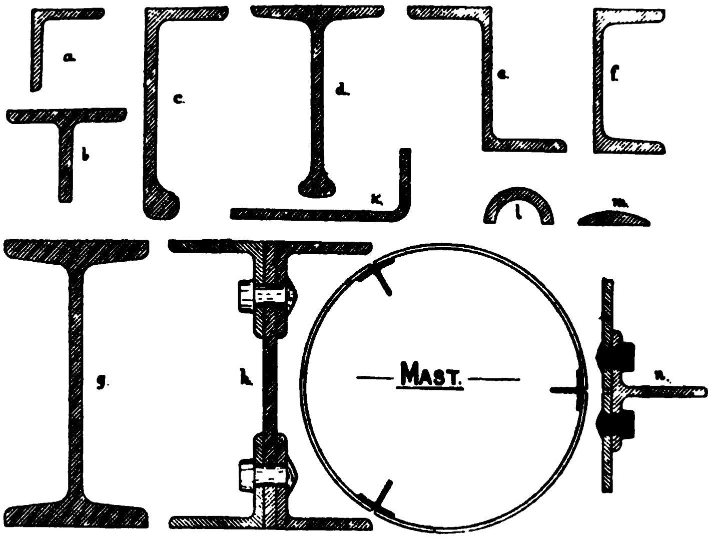


FIG. 8.

iron or steel for shipbuilding purposes is the possibility of obtaining the material in convenient and efficient forms. Thus, to form a beam in a wooden ship, it would be necessary to take a log of timber of solid rectangular section. We have seen in the previous chapter that in such a section the material is not disposed to the best advantage to resist bending. A beam of iron or steel can be rolled so that it has a broad upper flange and a bulb or flange at the lower part, both of which are so situated as to be most effective against bending. Examples are seen in Fig. 8.

The various sections used in Admiralty shipbuilding are shown in Fig. 8.

The *angle bar* (*a*) is used to connect plates together; as a stiffener to a plate; for beams, frames, etc.

The *tee bar* (*b*) is used as a stiffener to important bulkheads as (*n*) and inside masts.

The *channel bar* (*f*) is used as a stiffener in certain cases.

The *zed bar* (*e*) is largely used for transverse framing, to avoid the use of two angles riveted back to back. It is also used as a bulkhead stiffener.

The *I or H bar* (*g*) is used for the main stiffening of important bulkheads.

The *angle bulb* (*c*) is used for deck beams.

The *tee bulb* (*d*) is used for deck beams.

A *flange* (*k*) is frequently put on the edge of a plate to serve the purpose of an angle bar for stiffness or connection.

The *half round* (*l*) is usually hollow, and is used as a moulding round the ship.

The *segmental bar* (*m*) is used as a finish round the top of hatch coamings, etc.

The sections shown in Fig. 8 are drawn to scale, and show the exact form of sections now used. It is worth noting that the flanges of the zed, channel, and I bars are more substantial than the webs. These sections thus illustrate very clearly the principles touched upon in Chapter I. in connection with beams.

Rivets, Forms of, etc.—Fig. 9 shows the usual form of rivets and riveted connections employed in Admiralty shipbuilding.

A is the most common form of rivet, called a *pan head* rivet, from the shape of its head. It will be noticed that the rivet is formed with a conical neck. All rivets ½-in. diameter and above are thus formed, because the hole formed in the plate by punching has a slight taper, and it is most necessary that the rivet should completely fill the hole (see D, etc., Fig. 9).

D, E, F, Fig. 9, show various points associated with the pan head. D is the countersunk point necessary when the surface has to be flush, as for the skin plating. The hole formed by the punch has to be made conical by the countersinking drill. E is the point adopted for most of the internal work. No countersinking is done, and the point is hammered up full as shown. F is the point adopted where a finished appearance is desirable; this point is called a *snap point*.

B is a snap head rivet, and it is associated with a snap point, as G, when the riveting is done by the hydraulic riveter.

In some few cases it is necessary to have a flush surface on both sides, and in this case a countersunk head rivet is used, as C, and the point is also countersunk, as H.

Certain parts of the structure cannot be riveted by the ordinary means, as, for example, where outer bottom plating is connected to a stem casting. In such cases tap rivets are used, as L, M, and N (Fig. 9). In L, the most usual form, the head is countersunk, and the square projection is chipped off after screwing up. Where it is necessary to have the work portable, a recess is fitted in the head,

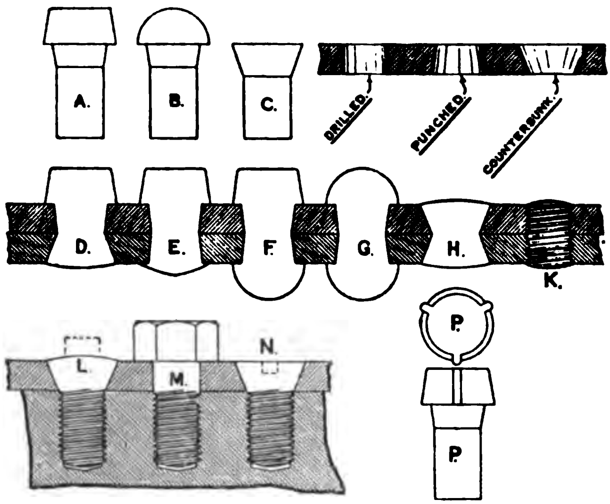


FIG. 9.

as N. Where a flush surface is not necessary, as for some internal work, the head is made hexagonal, as M.

In parts of the ship behind armour, the place would be dangerous in action, because when the armour is struck the rivet heads are likely to fly off. In such places, where men are likely to be, the inside of the framing is closed in with a steel lining (see Fig. 13, on the main deck). This lining is secured to the framing, etc., by screw rivets, as K, Fig. 9. Tap rivets, as L, M, or N, would be undesirable, because of the likelihood of the rivet breaking and the head flying out if the armour was struck.

Laps, Butts, etc.—When two plates are connected together,

they may either be *lapped* or *butted.* Laps are shown at F and G in Fig. 10. For thin plating a single row of rivets is sufficient, as F; for thicker plates a double row of rivets is necessary, as G. In the first case, the breadth of the lap, F, is three and a half times the diameter of the rivet used; in the second case, G, it is six times the diameter of rivet. This gives a clearance between the rows of $1\frac{1}{2}$ diameters, and rather more than a diameter clear of the edge. The edges of plates are usually lapped, but in some cases, where a flush surface is necessary, the edges are connected by an *edge strip.* A special form of edge connection is seen when the

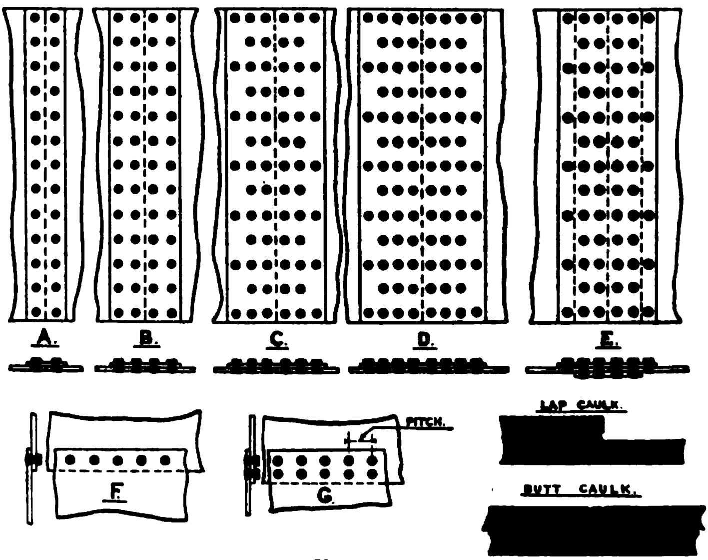


Fig. 10.

edge strip takes the form of a tee bar. Here the tee bar forms an edge strip and stiffener at the same time (see *n*, Fig. 8). When the end connection of plates has to be flush, as is usually the case, the connection is called a *butt-strap.* Butt-straps are single, double, treble, or quadruple riveted, according to the importance of the connection. These are shown by A, B, C, and D, Fig. 10, the breadths being respectively $6\frac{1}{2}$, $11\frac{1}{2}$, $16\frac{1}{2}$, and $21\frac{1}{2}$ times the diameter of the rivet used. Quadruple riveted straps are used for specially important joints (see Fig. 51 for one instance). In some cases butt-straps are made *double, i.e.* in two halves on either side.

Angle bars, etc., are connected together by a piece of angle,

fitted as shown in Fig. 14, long enough to take two or three rivets each side as necessary.

The breadths given above for butt-straps and laps are somewhat exceeded when dealing with high tensile steel. In this case a clearance of $1\frac{1}{2}$ diameters from the edge is considered necessary.

Spacing of Rivets.—The spacing of rivets from centre to centre along edges, butts, etc., is termed the *pitch.* This pitch varies according as it is necessary to have the joint watertight or not. For the former case the joint has to be *caulked,* and in order to do this it is necessary to have the rivets closely spaced to draw the work tightly together. The usual pitch for watertight work is from 4 to 5 diameters. Rather closer spacing, $3\frac{1}{2}$ to 4 diameters, is necessary for oiltight work. For non-watertight work a pitch of 7 to 8 diameters is all that is necessary.

Caulking.—All caulking should be metal to metal, filling pieces being avoided as far as possible. For laps the caulking edge must be made square (planed for important parts, as the outer bottom plating). The edge near the joint is nicked with a sharp tool and the piece so left is driven against the adjacent plate, as in Fig. 10. For butts the edges must be planed; a split is made either side of the joint and the two edges are forced together with a hollow tool, giving the shape to the butt, as in Fig. 10. Butt caulking is not so efficient as lap caulking, because a pull on the joint, or in-and-out working of the plating, is more liable to open the caulk in the former than in the latter case.

An interesting and very efficient form of butt-strap, shown at E, Fig. 10, is being adopted in some destroyers. The strap is double; the inside portion is treble riveted, and the outside portion double riveted. The alternate rivets are omitted in the last row, so that the plate is not weakened more than at the adjacent frame. The middle row is closely spaced to allow the edge to be lap caulked. If fracture of the plate occurred through this line of rivets it would be necessary to shear also all the rivets in the last row. The rivets in the two inner rows are in double shear.

CHAPTER III.

FRAMING OF VARIOUS TYPES OF SHIPS.

THE British Navy, intended to operate in all parts of the world, is necessarily made up of many different types of vessel. It would be obviously impossible in a work of this character to exhaustively

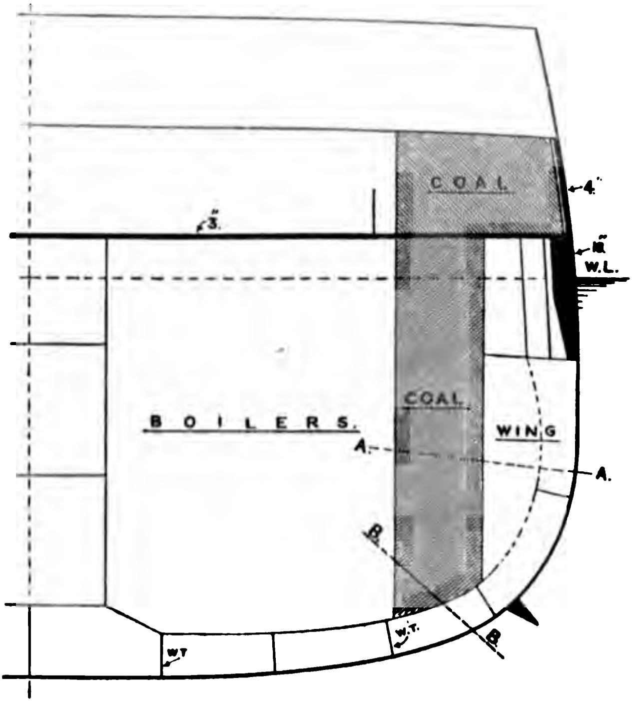


FIG. 11.—Section of H.M.S. *Royal Sovereign.*

describe the construction of each type of vessel in the Navy. All that is possible, or desirable, is to take certain main types, and deal with the principles of their construction. It will be seen that

the intended service of a ship has a distinct influence on the construction.

First Class Battle-ships.—These ships are heavily armoured and armed, and possess speed which is moderate compared with that obtained by cruisers. One distinctive feature of the construction of these ships is the provision of an inner skin up to the protective deck. Sections of recent battle-ships are shown in Figs. 12 and 13. The inner skin is $3\frac{1}{2}$ ft. in at the middle line, the depth being somewhat less up the bilge, and it is continued up to the protective deck in the form of a vertical bulkhead. There is, in addition, the inner coal-bunker bulkhead, so that at the side and bilge, at the lines AA and BB, there are three skins to pierce before a vital portion of the ship is reached; at the bottom there are only two skins to pierce. This arrangement is important, in view of the possibility of being rammed or receiving other damage under water.

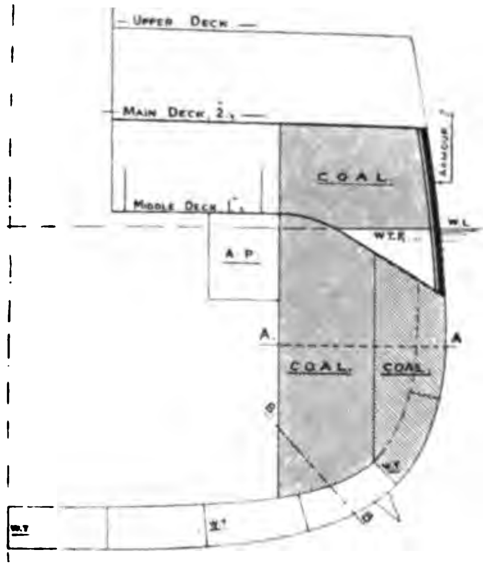


Fig. 12.—Section of H.M.S. *Duncan.*

The double-bottom arrangement is adopted in all ships of the Royal Navy above third class cruisers. A double bottom is valuable as providing an inner skin in the event of damage to the outer bottom, and it is always subdivided into a number of water-tight compartments, so as to localize any damage that might occur. A part of the space is conveniently arranged for the stowage of fresh water, forming the boiler *reserve feed.* Any of the double-bottom compartments, including the wings, can be flooded if desired to correct heel or trim caused by damage (see Chapter IX.).

In the *Royal Sovereign* the protective deck was level, with a thick belt, $8\frac{1}{2}$ ft. broad, as shown in Fig. 11. In the more recent

ships, the armour belt has been much reduced in thickness and increased in area, and the protective deck has been made level at the middle line, but sloping down to the lower edge of armour at the side (see Figs. 12 and 13).

Longitudinal Framing.—The main framing of these ships is arranged on the *longitudinal system*, the presence of the double bottom lending itself admirably to this arrangement. The fore-and-aft framing below protective deck over the length of the double

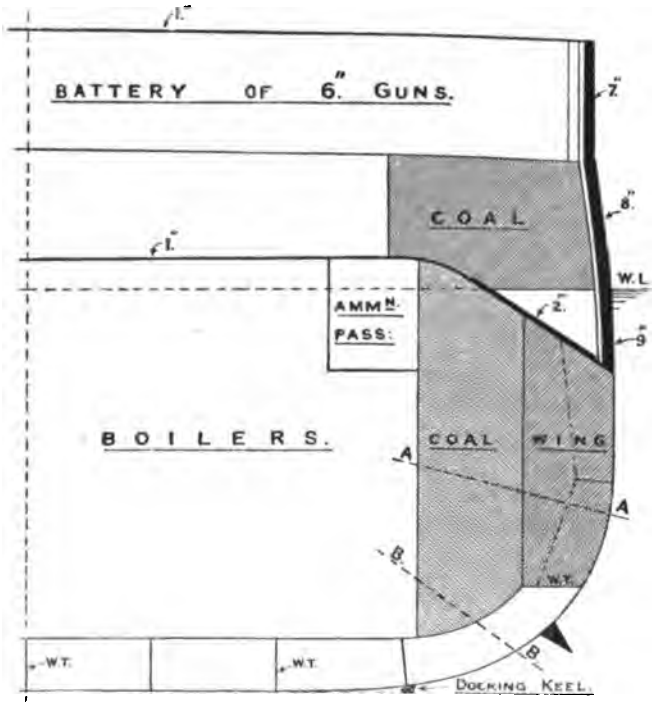


FIG. 13.—Section of H.M.S. *King Edward VII.*

bottom (about two-thirds the length) consists of a vertical keel, $3\frac{1}{2}$ ft. deep, and five longitudinal girders on each side (see Fig. 12). Of these the vertical keel, second and fourth longitudinals are water-tight, thus dividing the double bottom from side to side into six watertight compartments. The vertical keel and longitudinals are continuous, and the latter are allowed to taper somewhat in depth towards the ends of the double bottom. The vertical keel is 25 lbs. ($\frac{5}{8}$ in.) thick, with two angles along the bottom $4\frac{1}{2}$ in. $\times$ $4\frac{1}{2}$ in.,

and two along the top 4 in. × 4 in. These, together with the middle plate of the inner bottom, and the inner and outer plates of the flat keel, form a substantial *backbone* to the ship (see Fig. 3).

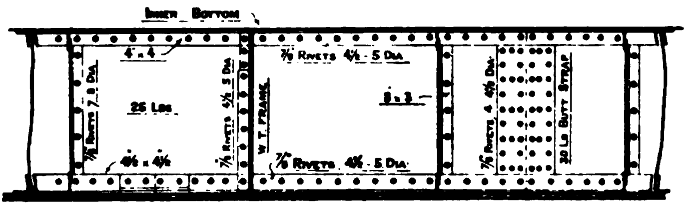


FIG. 14.—Watertight vertical keel.

The adjacent plates of the vertical keel are connected by 30-lb. ($\frac{3}{4}$ in.) treble riveted butt-straps (Fig. 14). Adjacent lengths of the angles at top and bottom are connected by bosom pieces of angle as shown. All the riveting is closely spaced for watertight work. The longitudinals are $17\frac{1}{2}$ lbs. ($\frac{7}{16}$ in.), with a single angle along the bottom 3 in. × $3\frac{1}{2}$ in., and a single angle along the top 3 in. × 3 in. The $3\frac{1}{2}$-in. flange is necessary to take the $\frac{7}{8}$-in. rivets used for the 25-lb. ($\frac{5}{8}$ in.) outer bottom, $\frac{3}{4}$-in. rivets being used elsewhere. The longitudinals are worked in this ship square to the outer bottom, but in a recent ship (Fig. 13) No. 4 is worked horizontal, to form a flat for the wing bunker. The arrangement of riveting depends on whether the longitudinal is watertight or not. The riveting in Nos. 2 and 4 is closely spaced (Fig. 15), and a double-riveted butt-strap is fitted. The connection of the ordinary frames only need the rivets spaced 7 to 8 diameters. Nos. 1, 3,

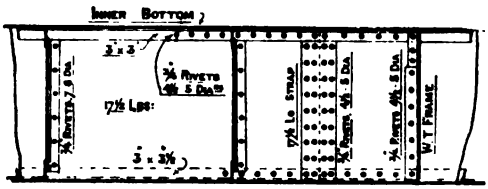
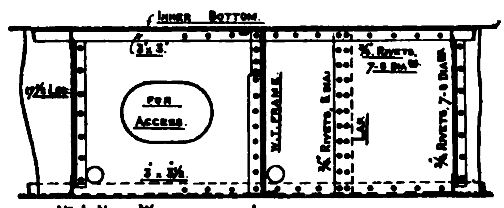


FIG. 15.—Watertight longitudinal.



—No. 1. NON WATERTIGHT LONGITUDINAL.—

FIG. 16.

and 5 have lapped butts (Fig. 16), and the riveting is more widely spaced. These longitudinals are lightened with manholes, which, in addition to lightening, serve the very necessary purpose of allowing freedom of access throughout the compartment. These non-watertight longitudinals also have holes at the bottom as shown, so that water will readily drain to the pump suctions.

Transverse Framing.—The framing running transversely

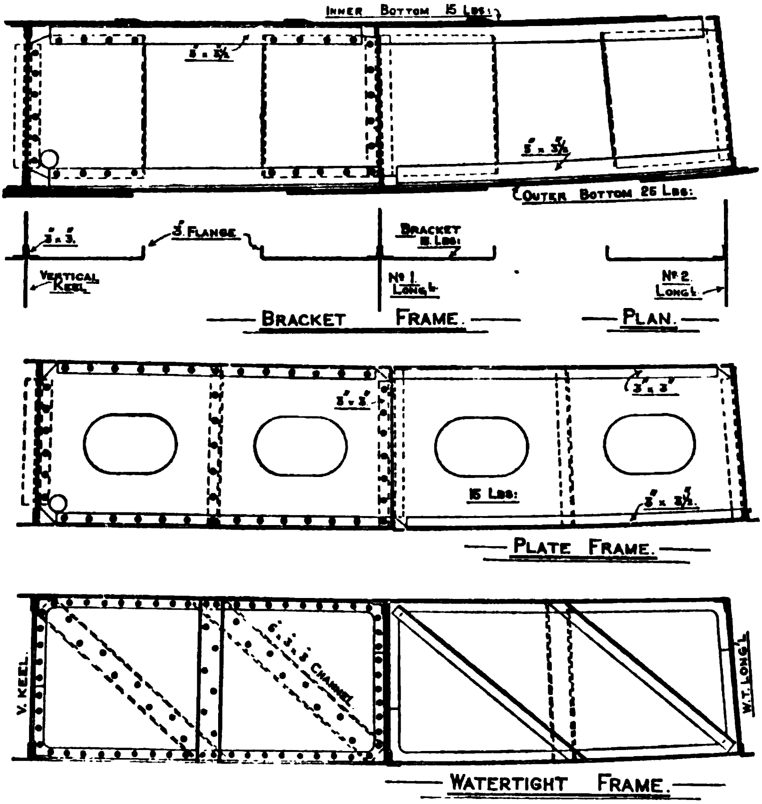


FIG. 17.

across the ship in the double bottom is worked intercostally, *i.e.* in short pieces between the longitudinals. Adjacent frames are generally 4 ft. apart. The frames are of three types, (i.) bracket frames, (ii.) solid plate frames lightened, and (iii.) watertight frames. Of these the second type is the most numerous, as it is the type adopted beneath the armour from No. 4 longitudinal

upwards, and under the heavy weights of the barbettes and the engines. The third class is fitted under the main transverse bulkheads (the inner bottom being continuous), and at intervals between (see Fig. 52). The bracket frames are fitted to the remainder. In one ship out of a total of 780, 456 were solid, 180 were watertight, and 144 were bracket frames.

The bracket frame is built up as shown in Fig. 17. Deep angles, 5 in. × 3½ in., are worked to the outer and inner bottoms, and to these in each bay are riveted two bracket plates, 15 lbs. (⅜ in.), with their inner edges stiffened by a 3-in. flange. These are connected to the vertical keel and longitudinals by pieces of angle 3 in. × 3 in.

The plate frame (Fig. 17) consists of a 15-lb. plate connected to the inner bottom, vertical keel, etc., by angles 3 in. × 3 in., and to the outer bottom by angles 3½ in. × 3 in. Each plate is stiffened by an angle bar, and holes large enough to enable a man to pass through for inspection purposes are cut to lighten the plate.

The watertight frames (Fig. 17) with the vertical keel and watertight longitudinals divide the double-bottom space into a large number of watertight compartments. The frame is made of 15-lb. (⅜-in.) plating under the transverse bulkheads, and 12½ lbs. ($\frac{5}{16}$ in.) in other places. The space is filled in solid with the plate, and staple angles are worked round the top and bottom as shown. This enables a tight fit to be made, and the whole is closely riveted for watertight work and caulked. These frames are specially stiffened, as shown, by three channel bars in the first bay, one channel and two angles in the other bays up to No. 4 longitudinal. This extensive stiffening is fitted to make the framing strong enough to stand the severe strains which exist when such a heavy ship is in dry dock. Recent ships are being fitted with docking keels at the side, as in Fig. 13. In these cases the stiffening of the watertight frames is not so extensive as described above.

Extra frames, both longitudinal and transverse, are worked under the engines in order to provide a rigid support.

Framing behind and above Armour.—The character of the framing behind armour is governed by the necessity of providing a rigid support to the armour. For armour 6 in. and 7 in. thick, the framing is formed of 10-in. zed bars, 24 in. apart, with fore-and-aft stiffening girders. For armour 9 in. thick, a more massive support

is necessary, and the framing is formed of plate frames 15 in. deep, 24 in. apart, with angles on the edges as shown in Fig. 18. There are fore-and-aft girders in addition, as shown. In either case the frames are well supported by bracket plates at the heads and heels.

The framing above the armour, between the main and upper

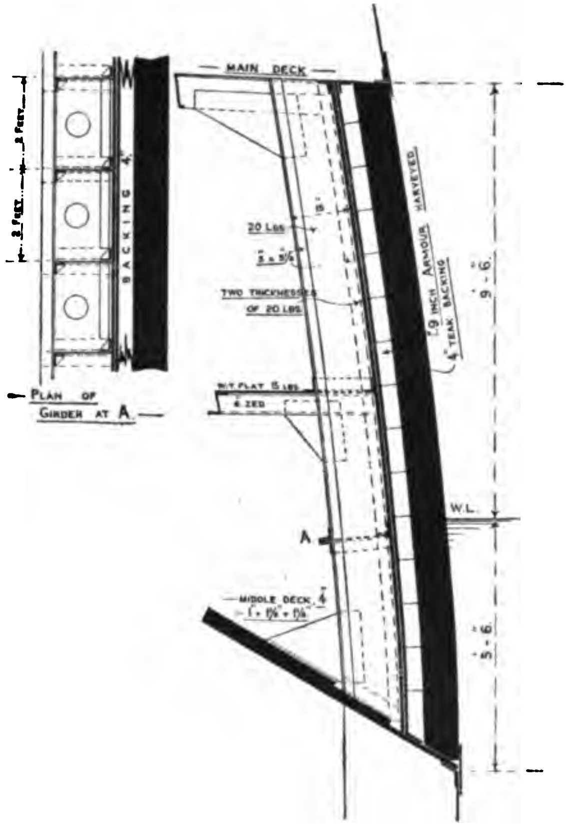


FIG. 18.—Support to 9-in. armour.

decks, consists of 6-in. zed bars 4 ft. apart, with 4 in. × 3 in. angles between. In ships with an armoured battery, as Fig. 13, the framing is made stronger to form a support for the armour.

Framing at the Ends of Ship.—The above description refers to the framing over the length of double bottom, or about two-thirds the length. The ends of the ship are framed on a somewhat

different principle. Here the longitudinal strength is of less importance, and the main function of the framing is to stiffen the outer bottom plating. Accordingly we find that the transverse framing is continuous either side of the vertical keel to the protective deck, and this framing is more closely spaced, viz. 3 ft.

The vertical keel is still continuous, but not watertight, and the frame consists of a vertical floor-plate with outer and inner angles to the bilge. Above this the frame is formed of a 6-in.

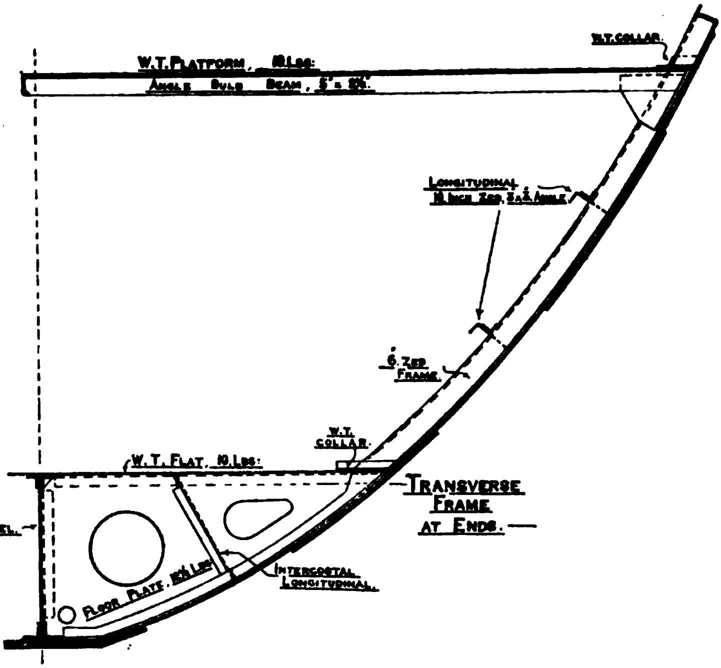


FIG. 19.

zed bar. In Fig. 19 the top of the floors is covered in with a watertight platform, and the zed bar is passed through with the inner flange cut away. This makes the watertight work at the ship's side more readily performed than if the complete zed bar went through (see Fig. 41).

As before stated, the vertical keel is continuous from end to end of the ship, but the five longitudinals on each side, which extend over the length of double bottom, are altered in character at the ends. They are either twisted round to connect on to a

fore-and-aft bulkhead or a flat, or tapered down to one of the forms shown in Fig. 19. The lower one is simply formed by an intercostal plate between each pair of floors, and the upper ones are each formed by a 10-in. zed, slotted over each frame, with a continuous 3 in. × 3 in. angle on the inside. Fig. 20 shows in detail how a longitudinal is tapered down over three frame spaces to the zed bar form. This is necessary to avoid any discontinuity in the fore-and-aft strength.

Above the protective deck the transverse framing is still 3 ft. apart, and is formed of 6-in. zed bars. In recent ships armour of varying thickness is fitted at the forward end, and behind this the frames are 2 ft. apart to well support the armour. At the extreme forward end, before the collision bulkhead, and below the protective deck, the frame consists of a solid plate connected

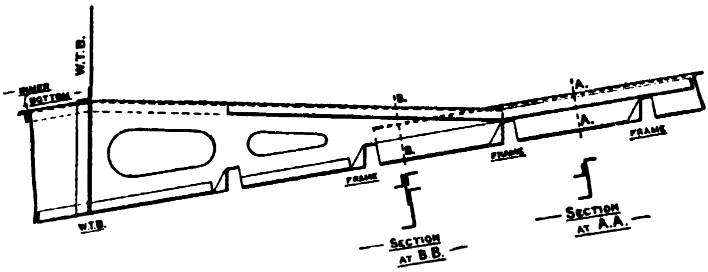


FIG. 20.

by angles to the outer bottom, etc. This plate is well lightened by holes.

It will be noticed that, although the inner bottom proper stops at about one-sixth the vessel's length from each end, yet an inner skin is obtained well towards the ends by the provision of the watertight flats forming the floors of the store-rooms, etc. (see Fig. 19).

First Class Cruisers.—Vessels of this type are in some cases of equal or even greater displacement than battle-ships, by displacement being meant the total weight of the ship. The broad distinction between a battle-ship and a first class cruiser of recent design, is that the former has thicker armour with a greater proportion of the side area protected and with a heavy armament of 12-in. guns, while the latter has lighter protection and

armament, but with high speed. The following comparison will illustrate this distinction:—

	Battle-ships.		First class cruisers.	
	Formidable.	*Duncan.*	*Drake.*	*Monmouth.*
Length	400 ft.	405 ft	500 ft.	440 ft.
Breadth	75 ft.	75½ ft.	71 ft.	66 ft.
Displacement in tons . . .	15,000	14,000	14,100	9800
Armour	9 in.	7 in.	6 in.	4 in.
Armament	4 12-in. 10 6-in.	4 12-in. 10 6-in.	2 9·2-in. 16 6-in.	14 6-in.
I.H.P.	15,000	18,000	30,000	22,000
Speed in knots, designed .	18	19	23	23

The design of first class cruisers has undergone great alteration in recent years. In vessels of *Edgar*, *Powerful*, and *Diadem* classes the protection was obtained by a thick protective deck near the waterline, in association with the stowage of coal above the deck (see Figs. 21 and 22). The great improvement in the quality of armour, brought about by the Krupp process, made it possible to armour the vessels of the *Cressy* class with a broad patch of 6-in. armour. This method of protection with thick main and middle decks has been a feature of first class cruiser designs since that time (see Fig. 23).

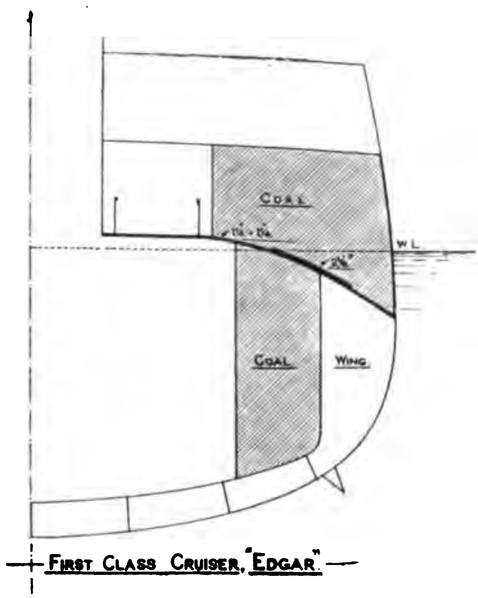


FIG. 21.

In the *Edgar* and *Diadem* (Figs. 21 and 22) the inner skin is continued up to the protective deck, but in later ships, as Fig. 23, the inner skin is only carried to the upper part of bilge. This gives a larger space

available for coal. In recent cruisers, also, the upper coal-bunker bulkhead has been dispensed with, to increase the facility of transporting the coal.

Many first class cruisers, including some of *Edgar* class and all the ships of *Diadem*, *Powerful*, and *Cressy* classes, have been constructed with the bottom sheathed with wood and copper. This has been done because these ships are intended to keep the sea for long periods without docking. The sheathing of copper is found to be the best anti-fouler. In building a ship, however, sheathed instead of unsheathed, certain sacrifices have to be made,

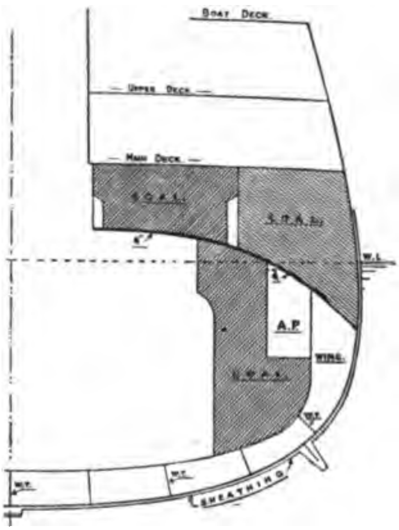


FIG. 22.—H.M.S. *Diadem* (sheathed).

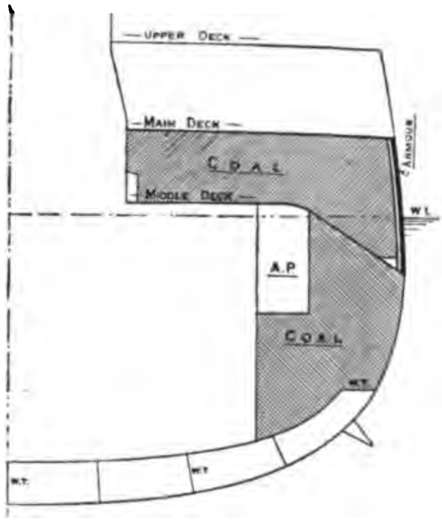


FIG. 23.—Armoured cruiser.

viz. increase of first cost and reduction of measured mile speed. We shall return to this subject later when dealing with methods of preventing fouling.

Framing of a First Class Cruiser.—Taking the framing of the cruiser whose section is shown in Fig. 23, we notice that the vertical keel is 42 ins. deep and 25 lbs. ($\frac{5}{8}$ in.). This depth is maintained over the whole length of the ship except in the engine-room, where the depth is about 5 ft. The two top angles are $3\frac{1}{2}$ in. $\times$ $3\frac{1}{2}$ in., and the lower angles $4\frac{1}{2}$ in. $\times$ $4\frac{1}{2}$ in. There are four longitudinals on each side, of which the second and fourth

are watertight. These extend over the length of the double bottom, *i.e.* about half length of ship.

The transverse framing in the double bottom is similar in character to that already described for a battle-ship. The framing behind the 4-in. armour is formed of 8-in. zed bars, with a fore-and-aft stiffening girder. The remainder of the framing, both longitudinal and transverse, is generally similar to that adopted in battle-ships as described above.

In this type of ship, in consequence of the great power of the engines, the framing in the engine-room is built exceedingly strong. We have noticed that here the vertical keel is 5 ft. in depth, and extra fore-and-aft girders are worked, in addition to the ordinary longitudinals, to give a rigid support to the engines.

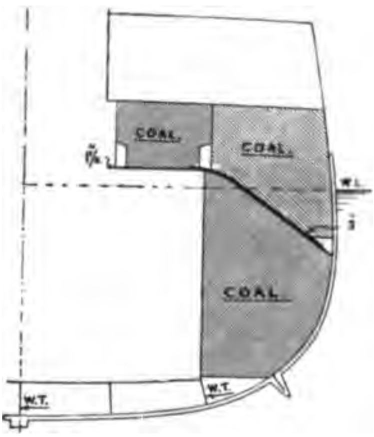


FIG. 24.—Sheathed second class cruiser.

Second Class Cruisers.—The majority of vessels of this type are sheathed with wood and copper, being intended for service on foreign stations, where the ships have to keep the sea for long periods. A typical section of a second class cruiser is given in Fig. 24. We notice that the protective deck and the coal above constitute the protection. The advantage of a double bottom is retained, but it is of less extent than in the previous cases considered, only extending to the coal-bunker bulkhead.

The construction of this ship is shown in Fig. 25. There is a continuous vertical keel 36 in. × 17½ lbs. ($\frac{7}{16}$ in.), with double angles along the top 3 in. × 3 in., and double angles along the bottom 3½ in. × 3½ in. There are two longitudinals on either side, 15 lbs. ($\frac{3}{8}$ in.), the second one being watertight. The double bottom extends over the length of the engine and boiler rooms, or rather less than half length.

For the transverse framing, the outer bar is 7 in. × 3 in., and is continuous from the keel to the protective deck, about 4 ft. apart. Within the double bottom, flanged bracket plates are worked as shown, with watertight frames and solid plate frames as

necessary. A large 10-lb. ($\frac{1}{4}$ in.) bracket is worked at the sides of the double bottom to ensure proper continuity of the transverse strength. Between the second longitudinal and the protective deck, a reverse bar, $2\frac{1}{2}$ in. × $2\frac{1}{2}$ in. (inadvertently omitted in Fig. 25), is worked on the back of the frame bar, and there is also a longitudinal girder, as shown. Above the protective deck the frame is formed by a 6-in. zed bar. The framing at the ends is

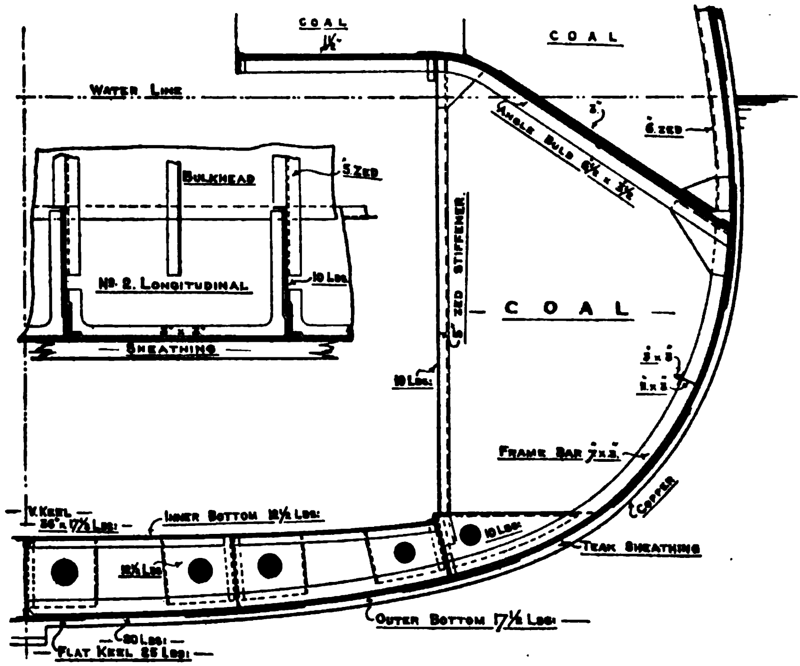


FIG. 25.—Framing of sheathed second class cruiser.

generally similar in character to that of battle-ships with floors, etc. This framing is worked every 3 ft.

The protection is obtained by a protective deck $1\frac{1}{2}$ in. on the flat, 3 in. on the slope, and the disposition of the coal is arranged to afford as much protection as possible.

The second class cruisers of the *Arrogant* class [1] are intended for use as *Fleet Cruisers*, and these vessels are not sheathed. The protection is obtained by a thick deck and side coal bunkers as usual. An extra longitudinal bulkhead is worked below at the

[1] *Arrogant, Vindictive, Furious, Gladiator.*

side, thus continuing the inner skin up to the protective deck (see Fig. 26).

Second class cruisers are the smallest vessels of the Royal Navy in which double bottoms are fitted, and in the remaining vessels we have to consider, the valuable element of safety provided by the inner skin has to be dispensed with.

Third Class Cruisers.—In these vessels we have a different set of conditions to consider. There is no double bottom, and the plating, being of small thickness, requires to be well stiffened. For such vessels, therefore, the adoption of the *longitudinal system* of framing would be unsuitable, and the best method is the *transverse system.*

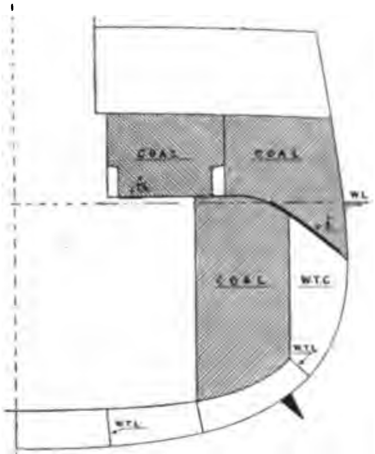


FIG. 26.—Section H.M.S. *Arrogant.*

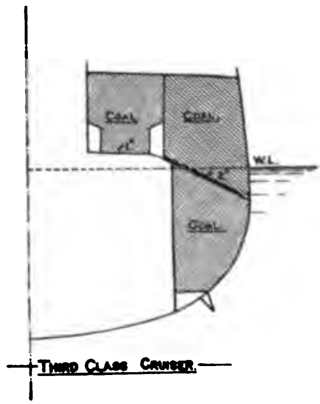


FIG. 27.

The section of such a ship is shown in Fig. 27, and the arrangement of framing is shown in some detail in Fig. 28. This framing is 24 in. apart throughout. The floor-plate, 10 lbs. ($\frac{1}{4}$ in.), extends from bilge to bilge. This plate is slotted out at the middle line to allow the lower bar of the *middle line keelson* to pass continuously through. The frame bar, 4 in. × 3 in., is continuous from the middle line to the protective deck on either side, and the reverse bar, 3 in. × 3 in., is continuous from the deck on one side to the deck on the other side, running along the top of the floor-plate as shown. Some difficulty is experienced in making the bunker bulkhead watertight where the frame, etc., pass through. In order to make a

satisfactory job, the bulkhead is stopped at the reverse bar, and a dished plate is fitted below between each pair of frames as shown. The frame above the protective deck is formed of a 4-in. zed bar.

The longitudinal framing consists of a middle line keelson and two side keelsons on either side. The middle line keelson is

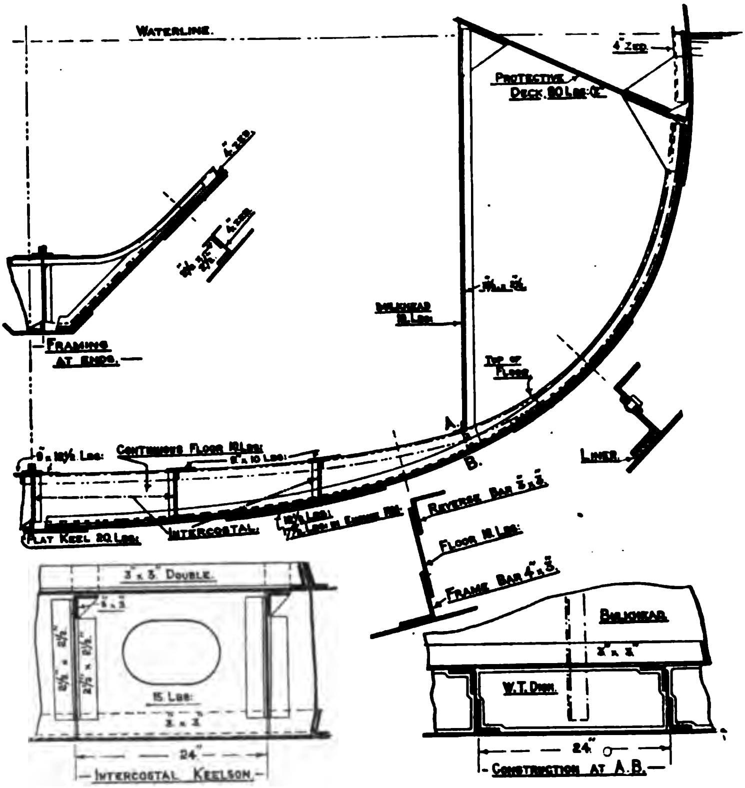


Fig. 28.—Framing third class cruiser.

formed by intercostal plates, 15 lbs. ($\frac{3}{8}$ in.), between adjacent frames, connected to the floor-plates by angles $2\frac{1}{2}$ in. × $2\frac{1}{2}$ in. These plates project above the floors and connect to two continuous angles, 3 in. × 3 in. The lower edges of the intercostal plates connect to a continuous angle, 3 in. × 3 in. There are in addition two continuous *rider* plates, each 9 in. × $12\frac{1}{2}$ lbs. ($\frac{5}{16}$ in.), running along the top of the floors. The side keelsons are

intercostal, with a rider plate on top as shown. Below the protective deck, before and abaft the machinery space, the frame is formed as shown in Fig. 28, with a 4-in. zed bar connecting on to a 10-lb. ($\frac{1}{4}$ in.) floor-plate. Above the protective deck the frame is formed of a 4-in. zed bar.

Sloops.—A large number of small vessels, about 1000 tons displacement, called *sloops*, are employed on foreign stations. They carry some sail-power, and are sheathed with wood and copper for the reasons given above.

A section of such a ship is given in Fig. 29. One feature of these vessels is the absence of any protective deck. In the side bunkers, however, there is a division, at about the level of the waterline. It is intended that the coal should remain in the upper part of the bunker as long as possible, in order to serve the purpose of protection, and to assist in preserving the stability if the side were pierced.

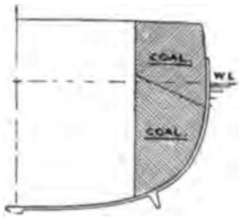


Fig. 29.—Sheathed sloop.

These ships are framed on the transverse system with the frames 24 in. apart. Below the watertight division in the bunkers a frame bar, 4 in. × 3 in. (Fig. 30), is worked from side to side. Between the coal-bunker bulkheads a floor-plate is worked, 10 lbs. ($\frac{1}{4}$ in.), with a reverse bar, 3 in. × $2\frac{1}{2}$ in., on the upper edge. A 3 in. × $2\frac{1}{2}$ in. reverse bar is connected to the frame bar above the bulkhead. Above the bunker division the frame consists of a 4-in. zed bar. It will be noticed that the bunker division severs the transverse frame completely, and to maintain the continuity of the transverse strength bracket plates are worked as shown.

The middle line keelson (Fig. 30) is formed by an intercostal plate, 15 lbs. ($\frac{3}{8}$ in.), between each pair of frames, with staple angles, 3 in. × 3 in., connecting to the flat keel and to the floors. The intercostal plates project above the floors and connect to two continuous angles, 3 in. × 3 in. An intercostal keelson is also worked on either side.

Torpedo-boat Destroyers.—The essential feature of this type of vessel is speed, and every effort is made by careful design, high quality material and careful workmanship, to keep down the weight of the hull structure to the lowest amount possible. Sections of a typical destroyer are given in Figs. 31 and 32 the

first showing the transverse portions of structure, and the second those portions which give longitudinal strength.

The ship is framed on the transverse system with a frame, floor, and reverse frame. Deep frames are fitted at intervals. There is a middle line intercostal keelson with continuous angles at the top and bottom. The engine and boiler bearers are so arranged as to assist materially in providing longitudinal strength

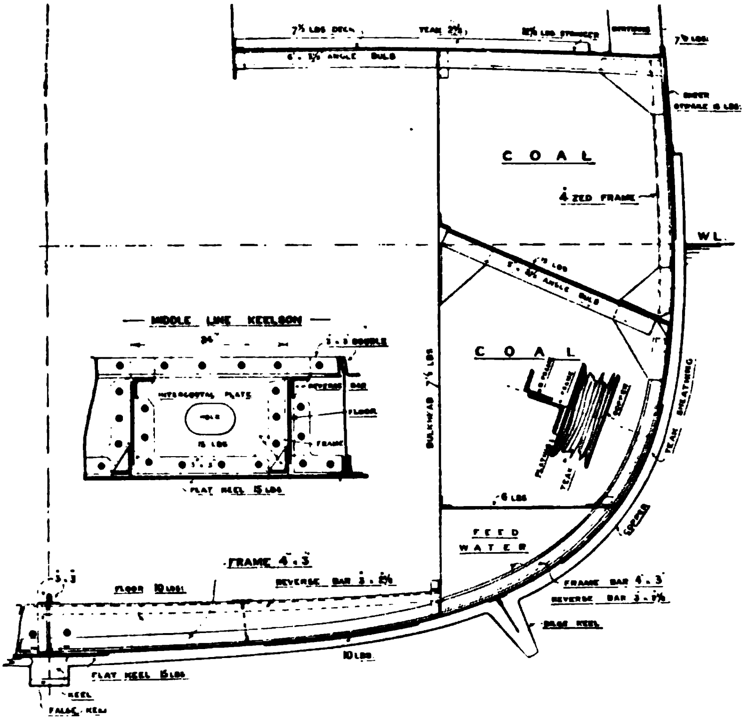


FIG. 30.—Midship section sheathed sloop.

to the structure. Flanging of plates is largely adopted to save the weight of connecting angle bars. The deck is of special importance, as it is severely strained if the ship is subjected to a *sagging* moment. In this case the deck between the beams is liable to buckle, being of thin plating, and in order to enable it to effectively stand the strains, it is well stiffened by fore-and-aft girders. Deep beams are fitted at intervals.

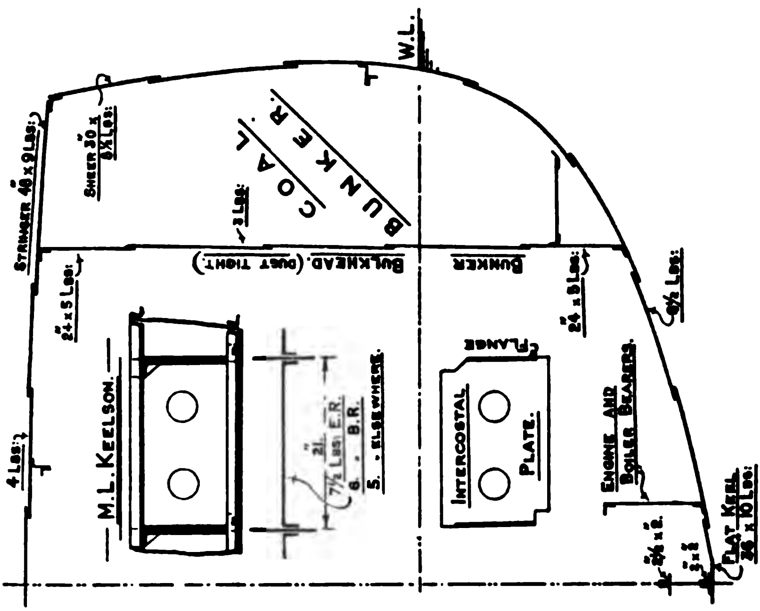


FIG. 32.—Fore-and-aft structure of a destroyer.

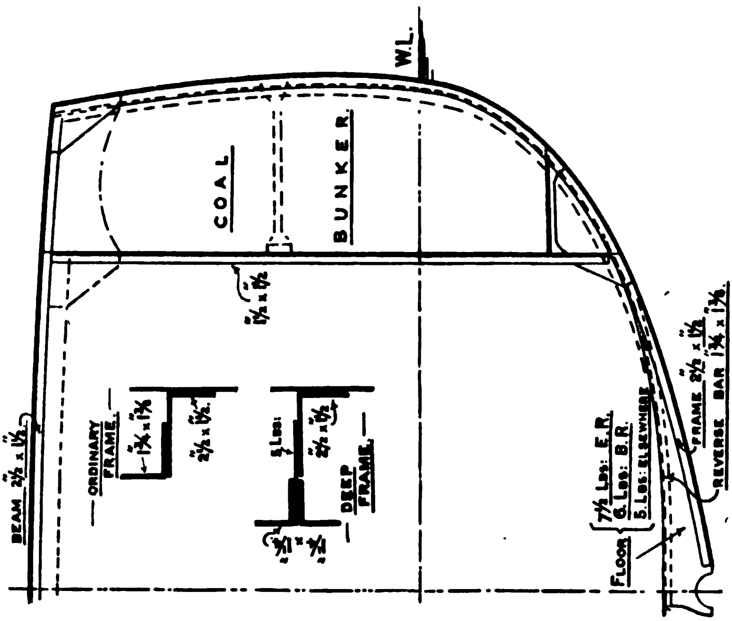


FIG. 31.—Transverse structure of a destroyer.

CHAPTER IV.

BEAMS, PILLARS, AND DECKS.

Beams.—The transverse framing we have been considering ends at the upper deck. To complete the transverse structure we have *beams* connecting the sides of the ship together at the level of the various decks and platforms. Beams not only tie the sides of the ship together, but they form the support to the decks and

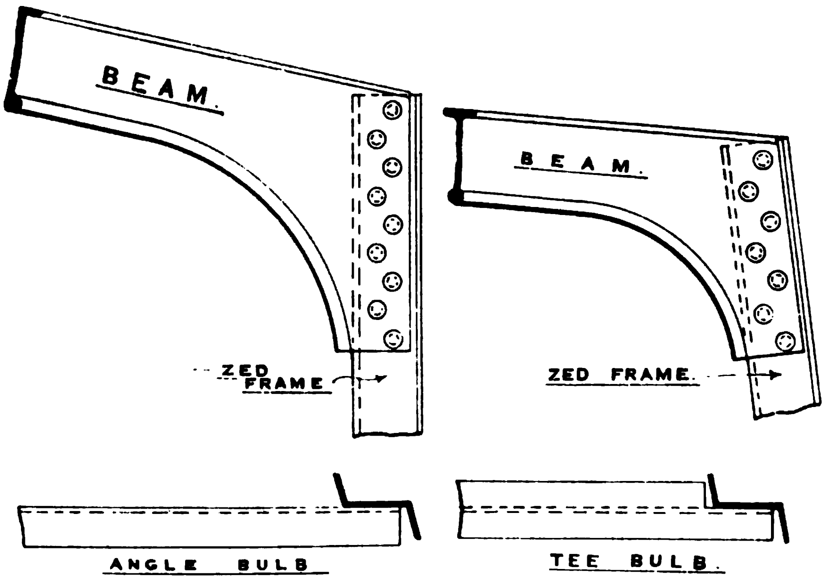


FIG. 33.

platforms. Beams for decks on to which water is likely to come, as the upper and main decks, are made with a *round down* in order that the water may run to the side scuppers. The amount of this round is 9 in. in a 75-ft. battle-ship, and 6 in. in a 40-ft. cruiser. Beams to the lower protective decks are of

the same shape as the deck, usually level at the middle line and sloping down to the sides (see Fig. 12). Beams to the lower platforms and decks are level (see Fig. 19).

Beams in a large ship are spaced every 4 ft. where the frame spacing is 4 ft., and every 3 ft. at the ends of the ship. For a small cruiser the beams are placed on alternate frames, *i.e.* every 4 ft.

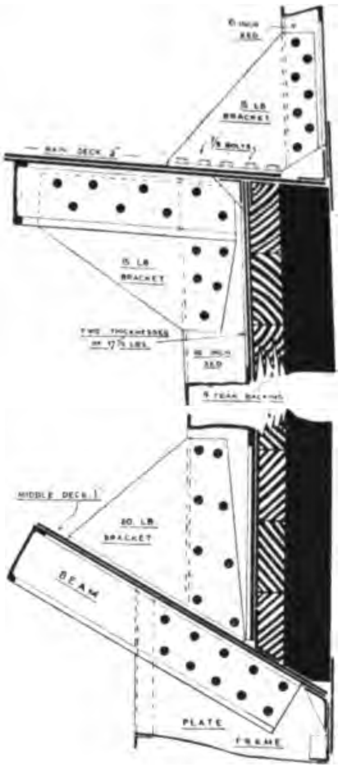


FIG. 34.

Beams are most commonly formed of angle bulb (*c*, Fig. 8). Most decks are now covered with steel plating, and the angle bulb is then a convenient beam to use. When, however, a wood deck has to be laid direct on to the beams, as is sometimes the case, it is more desirable to have the tee bulb (*d*, Fig. 8), in order that the deck bolts may be worked zig-zag, and not in a direct line, as would be the case with the angle bulb. The tee bulb is a convenient form to use for skid beams for supporting the boats. A zed bar (*e*, Fig. 8) is a convenient form of beam when the flat supported forms the crown of a magazine in which teak lining is fitted. The lining can be bolted to the inner flange of the zed. In recent ships, however, the lining to magazines has been dispensed with, so that this form of beam is not necessary. Angle bars are used as beams to flats in which the greater strength of the angle bulb is not required.

The connection of beams to the transverse frames is of great importance, as this, together with the transverse bulkheads, helps to prevent the racking of the ship due to rolling. To ensure an efficient connection, the beam is connected to the frame either by a beam arm, or a bracket plate. The beam arm is used where a neat appearance is desirable, as below the upper deck. To form the beam arm, the beam is cut at the middle of the

web and the lower part is bent down. A piece of plate is then welded in, giving the form shown in Fig. 33. The bracket is used in places where a neat appearance is not so desirable (Figs. 25 and 30). The usual depth of the beam arm or bracket is two and a half times the depth of the beam, so as to get a good riveted connection to the frame. In the special case of the beam to the middle deck, a bracket is not necessary, as sufficient rivets are obtained through the solid plate frame, which is worked beneath the armour (Fig. 34).

Half-beams and Carlings.—Some beams come in way of openings in the deck, as the engine hatch, ventilators, funnel

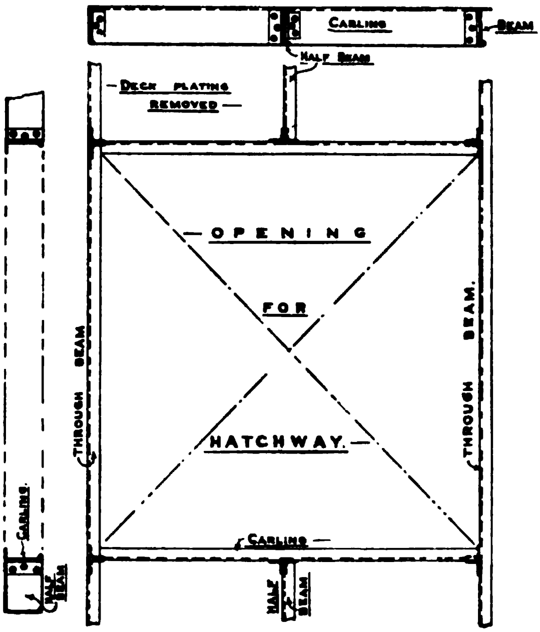


FIG. 35.

casings, etc. In these cases it is necessary to cut the beams; these are then termed *half-beams.* The inner ends of the half-beams are connected to a fore-and-aft *carling,* which extends to the first complete beam at each end of the opening. Fig. 35 shows this arrangement for a small opening for a hatchway. The carling in this case is formed of an angle bulb of the same size as the

beams. It will be noticed how the half-beam hangs on the carling, which in its turn hangs on the through beams at the ends.

At large openings the carling is made exceptionally strong, because of the large number of half-beams hanging on to it. In Fig. 39 it is formed of a deep 20-lb. ($\frac{1}{2}$ in.) plate, with an angle bulb in addition. The deep plate also provides a convenient attachment for the thin funnel casing. The specially large openings in the protective deck caused by the spread of the funnels as they go down to the boilers cut off such a large number of beams that some compensation has to be provided. This is arranged for by running across at intervals, where possible, strong beams, usually built up as (*h*), Fig. 8.

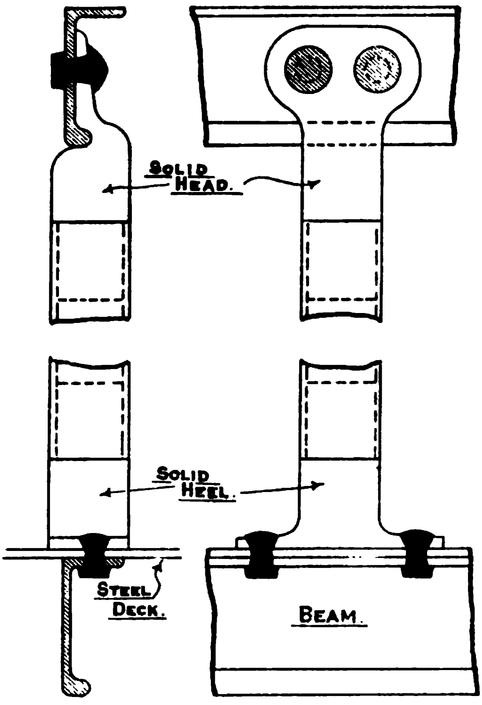


FIG. 36.

The upper deck, forward and aft, needs to be strengthened in way of the blast of the heavy guns. This is arranged for by working fore-and-aft girders, well supported by pillars, under the upper deck.

Pillars.—It is essential that the long beams in a ship should be supported at other places than at their ends, as, by thus supporting them, their capacity for bearing a load is greatly increased. In ships of the Royal Navy nearly every beam is supported by pillars or in some other way. Where possible, the fore-and-aft bulkheads are arranged so that the stiffening bars to the bulkhead act also as supports to the beams. One instance of this may be noticed in the cabin bulkheads below the upper deck aft in a battle-ship. In other places pillars are fitted. These pillars are

made of wrought steel tubes, a usual size for the 'tween-decks being 5 in. diameter and ¼ in. thick. In boiler-rooms and under

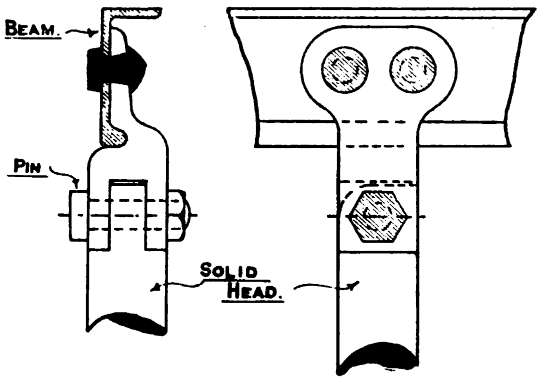


FIG. 37.

barbettes, where the pillars are very long, the diameter is 10 in. and the thickness ¾ in.

A considerable saving of weight results from using hollow pillars instead of solid, as well as the advantage of the greater stiffness of the hollow form of larger diameter. Thus, suppose a pillar if hollow could be 6 in. diameter and ½ in. thick, and if solid 4¾ in. diameter (these two pillars in merchant shipbuilding would be considered of equal strength), the saving of weight by using the hollow form would be over a ton for every 100 ft. of pillar worked into the ship.

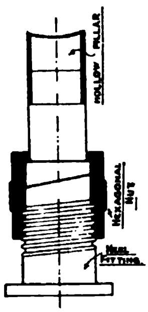


FIG. 38. — Heel fitting to portable pillar.

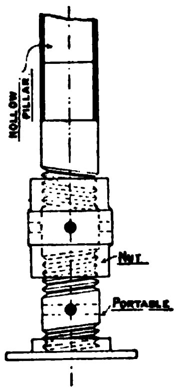


FIG. 38A. — Heel fitting to portable pillar.

Hollow pillars must have solid heads and heels, and they must be secured at the top and bottom so as to form a *tie* between the decks as well as a *strut* (Fig. 36). Pillars are so arranged as to form continuous vertical lines of support right down to the bottom of the ship. The heel of the pillar should be riveted direct to the beam, and not bolted to the wood deck (if fitted). This latter may be done when the

pillar supports the skid beams, and is thus not fitted to assist the structural strength of the ship.

In many cases where pillars are necessary, the obstruction caused is very inconvenient, as in way of capstan gear, torpedo-tubes, etc. In such cases the pillars are made portable. The head is fitted with a pin (Fig. 37), so that the pillar can be hinged up clear, or, if desired, removed altogether for the time. Figs. 38 and 38A show two forms of heel fitting; in the first case there is an inconvenient obstruction left, in the second this heel fitting is also made portable.

Deck Plating.—Plating is worked on decks and flats for various reasons, (*a*) for the purpose of contributing to the *structural strength*, as on the upper deck; (*b*) for the purpose of *protection*, as on the main and middle decks in Fig. 12; (*c*) to divide the ship into *watertight compartments*, as in the hold forward and aft; (*d*) to distribute the strains on the deck, as under bollards, boat-hoists, etc.

The Upper Deck is a most important part of the structure, especially in a vessel of large proportion of length to depth as a cruiser. It forms the upper flange of the girder, and so contributes materially to the ship's structural strength. Fig. 39 gives the arrangement of the upper deck plating amidships in such a ship The *side stringer* has a total breadth of 10 ft., worked in two strakes of 20-lb. (½ in.) high tensile steel. There is also a *funnel stringer* 5 ft. wide, also of 20-lb. high tensile steel. This runs fore and aft along the funnel and engine hatches as shown. The remainder of the plating is 10-lb. (¼ in.) mild steel. The plating of this deck is laid direct on to the beams, with single riveted edge strips on the upper side connecting the edges together. The butts are all double riveted; those of the 20-lb. plating being *double*, *i.e.* on both sides of the plate. Holes that are necessary in the deck for coal shoots, hand-ups, etc., are carefully compensated for by pieces of plate on either side of the hole to make up the sectional area cut away. The butts of the various strakes of plating are carefully shifted clear of one another, and clear of the butts of the sheer strake (see Chapter V.).

The above instance has been taken to show the care taken to render the upper deck specially strong when such strength is required, but in many cases, where the proportion of length to depth is not so great, this exceptional strength is not necessary. In such ships, as in *Majestic*, the plating does not completely cover

the deck, the most important part being the stringer next the side. We also find that only such a stringer is worked at the ends of many ships.

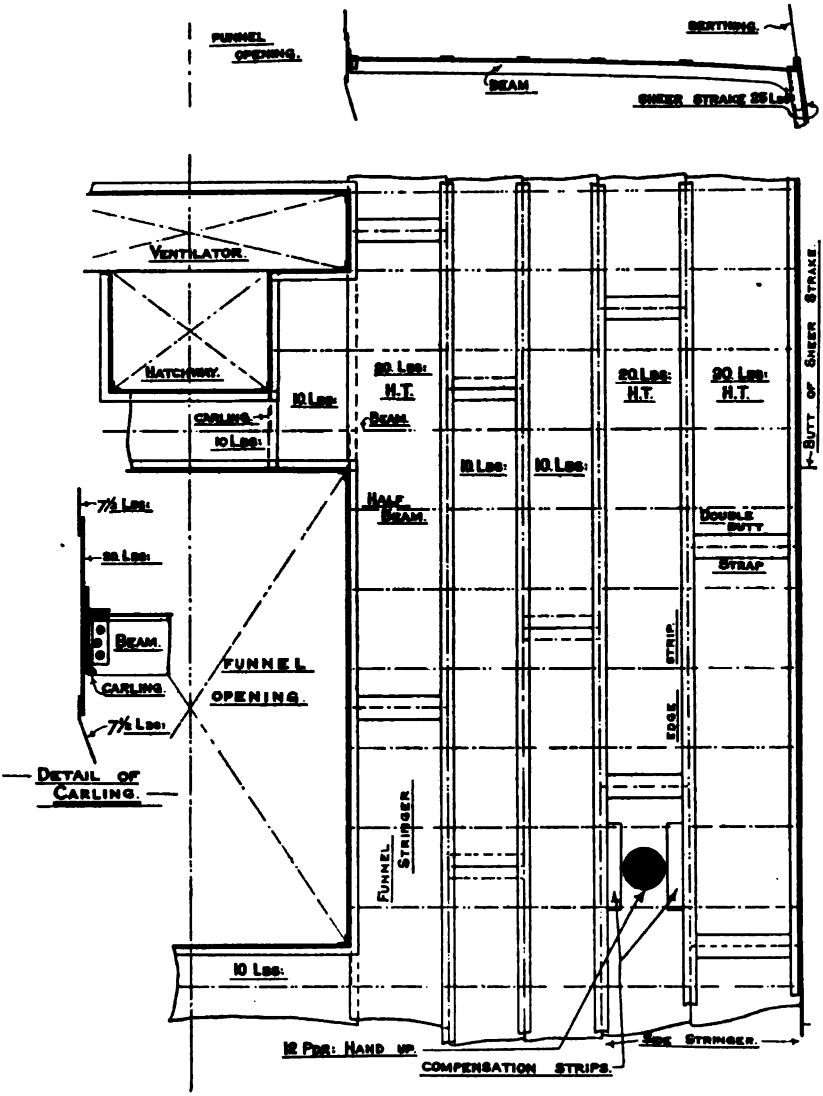


FIG. 39.—Upper deck plating of cruiser.

In the most recent ships, however, decks are nearly always

completely plated; when wood is worked above as for the upper deck, there is less likelihood of the wood catching fire with the steel underneath. For other decks we have *corticine*, and this necessitates a complete steel deck to receive it.

In the most recent large ships an armoured battery is worked between the main and upper decks, and this is covered in by a protective deck, about 1 in. thick. This deck is very advantageously situated to assist in the ship's structural strength.

Protective Decks.—In decks fitted for the purpose of protection the plating is usually worked in two or more thicknesses. Thus in the *Duncan* the main deck is 2 in., worked in two thicknesses of 40 lbs. The middle deck is 1 in., and here two 20-lb. plates are worked. In this way no edge strips or butt-straps are necessary, as each thickness acts as security for the edges and butts of the other thickness. The lower thickness only is riveted to the beams; the general arrangement of the riveting is shown in Fig. 40.

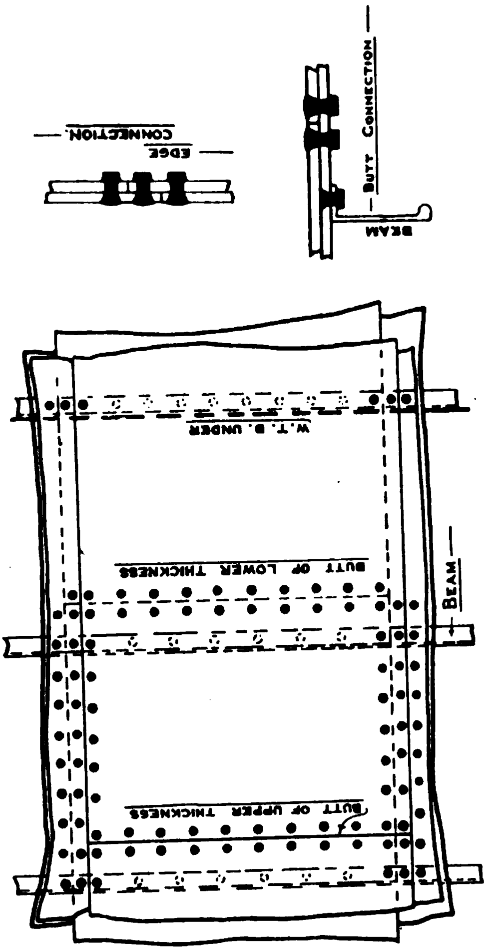


Fig. 40.—Riveting in protective deck.

Platforms, etc.—The plating of platforms and flats is usually of 10-lb. ($\frac{1}{4}$ in.) plating, with both edges and butts lapped and single riveted. For compartments where a good foothold is desirable, and where there is a lot of heavy wear and tear, the

flat is formed of 15-lb. galvanized *ribbed* plating. In a large ship the flats of the submerged torpedo-rooms, auxiliary machinery compartment, etc., are fitted in this way.

Watertightness of Flats, etc.—In watertight platforms, etc., the connection of the plating with the ship's side must be made watertight. Fig. 41 shows two methods of doing this, one when the zed bar frame passes through, and the other when a simple angle bar passes through. In either case angle bars are smithed round to cover the holes completely; the riveting is closely spaced and the whole carefully caulked and made watertight. The pressure such a flat, 20 ft. below water, would have to stand if the compartment below were bilged amounts to about 60 tons per

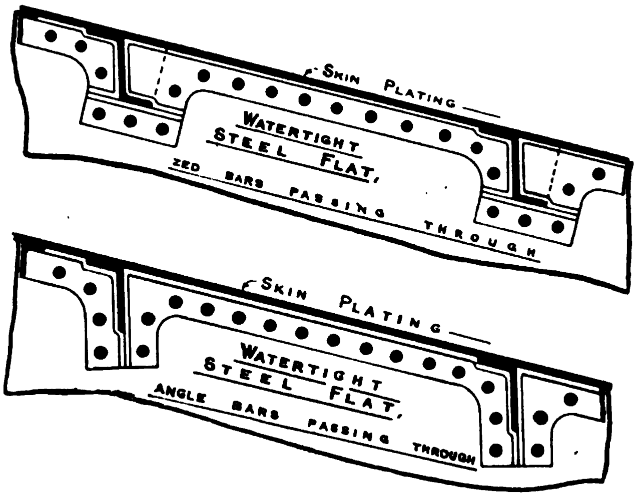


FIG. 41.

100 square ft., so that substantial work is seen to be very necessary. There is no difficulty in securing a watertight connection with the side at the main or middle decks of an armoured ship, because the transverse framing is not continuous through the deck, and a fore-and-aft angle can be run along and caulked (Fig. 34).

Wood Decks.—The use of wood decks has been very much reduced in recent years in vessels of the Royal Navy, on account of the probability of the wood catching fire in action. Wood is now only used for weather decks, in the Admiral's apartments and the ward room, in casemates and ammunition passages, and for magazine flats. In the Admiral's apartments and ward room

the flat is of Dantzic fir, 2 in. thick. In the other cases the flat is of teak, on account of the suitability of this timber to stand heavy wear and tear. For the weather decks the thickness is generally 3 in., with thicker planks called *waterways* round the edges of the decks, forward and aft, and round barbettes, etc. Thicker planks are worked in way of the rub of chain cables. In casemates the teak is 2 in., and in ammunition passages $1\frac{1}{2}$ in.

When a steel deck is laid, the wood deck is fastened by

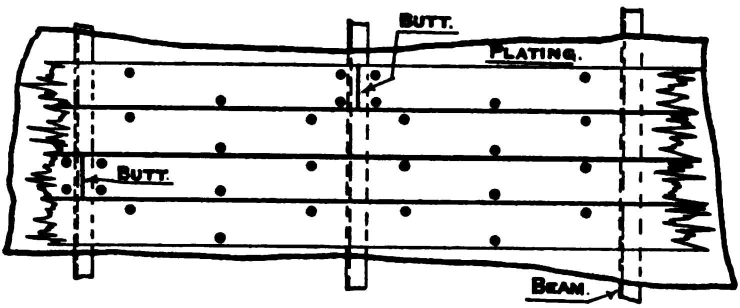


FIG. 42A.

through bolts to the plating between the beams. To decks which have to stand severe compressive strains, the connection of the wood deck to the steel deck is made a very efficient one, so that the wood and the steel may act together in resisting the buckling (see Fig. 42A).

When, however, as is sometimes the case, the wood deck has

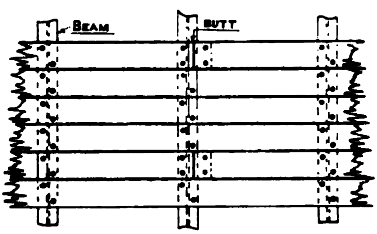


FIG. 42B.

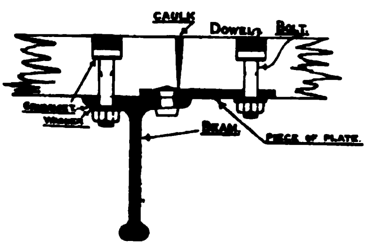


FIG. 43.

to be laid direct on to the beams, the bolts must be taken through the upper flange of the beam. For this reason, when a wood deck is thus worked, it is desirable to make the beam of tee bulb, so

that the bolts may be placed zigzag, as in Fig. 42B, and not in one continuous line, as would be necessary with an angle bulb beam. When thus laid direct on the beam the bolts of the butt of deck plank would be too close if taken through the flange of the beam, and at each butt, therefore, a short piece of plate, having the width of the plank, is fitted to take the bolts (Figs. 42B and 43).

Planking is fastened to the steel deck or the beams by means of galvanized iron bolts with round heads, as Fig. 44. The heads are let in well below the surface of the deck, and the cavity is filled in with a wood plug called a *dowel*, well steeped in white lead. A hempen grummet is placed under the head, and the nut underneath has a plate washer and grummet (Fig. 43). All this is necessary to make the hole in the deck properly tight. It will be noticed that the neck of the bolt is square to prevent the bolt turning when the nut is being screwed up.

The edges and butts of deck planking are caulked to make

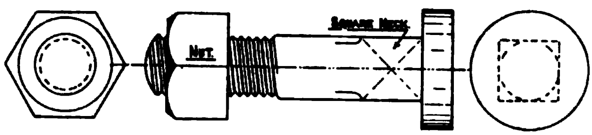


FIG. 44.

the deck watertight. The edges and butts are left with a slight opening at the surface, as Fig. 43, and into this oakum is forced, filling the opening right down to the bottom of the seam. The top is then payed with pitch. When carefully done this caulking should make the deck properly tight.

Corticine.—In the living spaces of the ship other than those mentioned above, no wood decks are fitted, but *corticine*, a thick linoleum, is laid direct on the surface of the steel deck. The steel deck has to be laid flush for this purpose, with edge strips on the under side worked between the beams. The upper deck of torpedo-boat destroyers is also laid with corticine.

The corticine is secured to the deck by means of a solution of orange shellac and methylated spirit, and when secured the edges are stopped with putty. Round scuttles and exposed edges the corticine is secured in addition to the deck by galvanized iron strips screwed into the plating. For thin decks, as in destroyers,

these strips are secured to the plating by bolts with a nut below having a washer and grummet. Where heavy wear takes place on a steel deck covered with corticine, as at the foot of ladders, the corticine is covered with plates of ribbed iron.

Watertight Hatches.—The hatches to ordinary flats and decks are fitted with a raised coaming and a steel cover with an indiarubber joint. This indiarubber should be periodically examined, and where perished should be renewed, as a defective cover may destroy the watertightness of a complete deck. Scuttles and armour gratings to thick decks are dealt with in Chapter XIII.

CHAPTER V.

PLATING OF THE OUTER AND INNER BOTTOMS.

Outer Bottom Plating.—The weight of this plating forms a good proportion of the total weight of the hull structure. It is a most important portion of the structure, because it not only contributes largely to the structural strength, but it keeps the ship watertight. One advantage of forming this plating of *mild steel* has already been referred to, viz. the advantage of the ductility of steel as compared with iron. Steel ships have frequently grounded without making water, under circumstances in which an iron ship would have been in a serious condition owing to the rupture of the plating.

Shift of Butts.—An important point in connection with the longitudinal structure of any ship is the arrangement of a good

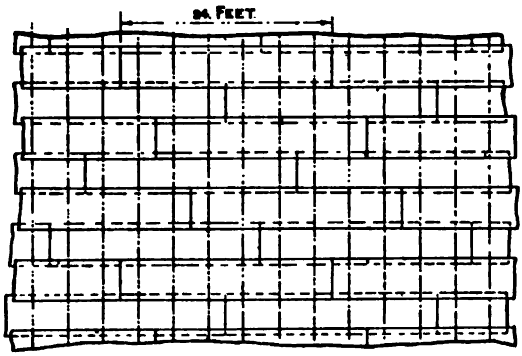


FIG. 45.—Shift of butts.

shift of butts. A butt must be a place of relative weakness, and the butts of the various portions of the fore-and-aft structure are arranged well clear of each other. Thus, for the outer bottom plating, it is laid down that butts are not to be closer together in

the same frame space than two passing strakes. With plates as

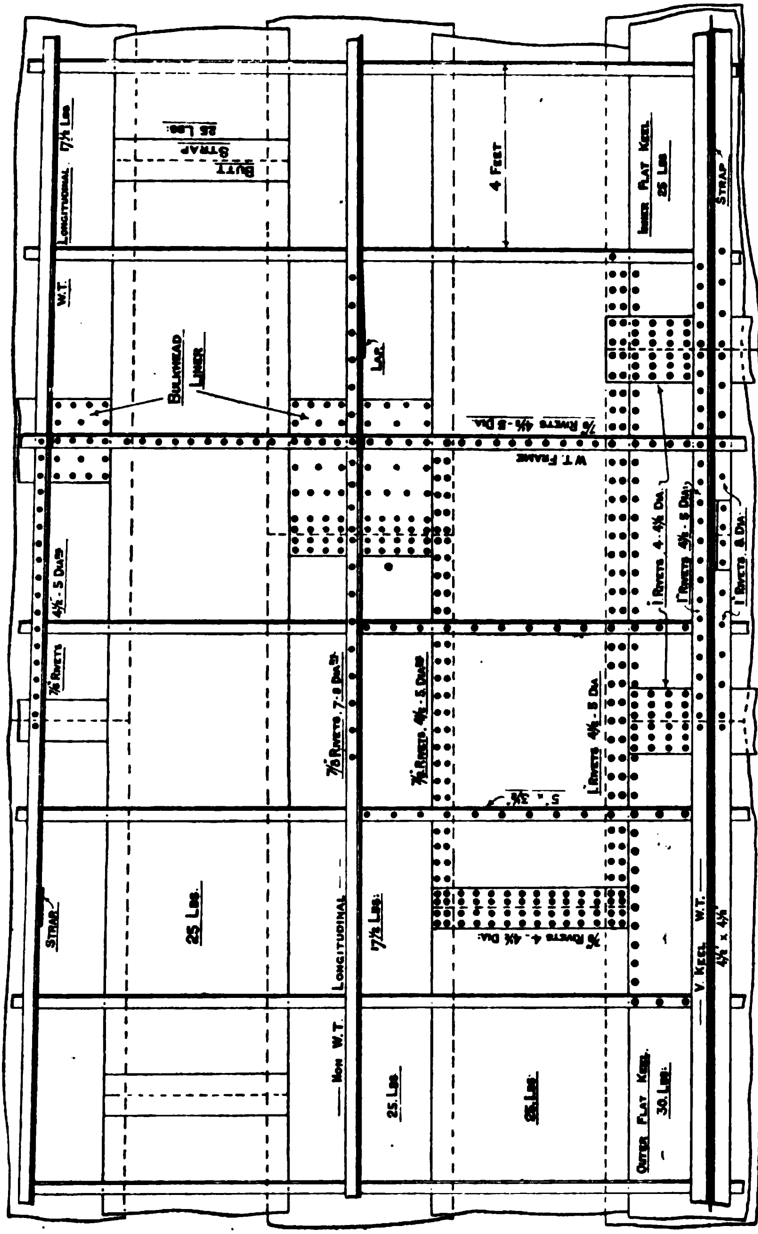


FIG. 46.—Riveting, etc., in outer bottom plating.

now worked, 20 ft. in length and over, there is no difficulty in considerably exceeding this condition. Thus, in Fig. 45, 24-ft. plates are worked and five passing strakes are obtained. The same principle has to be borne in mind when arranging the butts of the vertical keel and longitudinals, inner bottom, etc., so as to avoid any weakness of the structure in any one frame space (see Fig. 46).

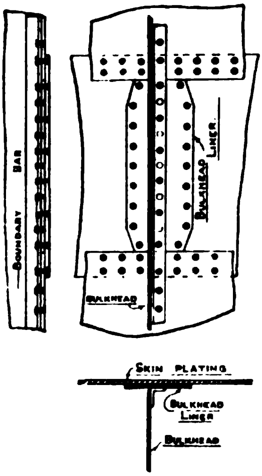


FIG. 46A.—Special form of bulkhead liner.

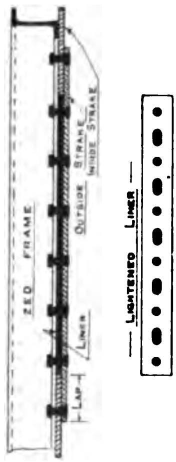


FIG. 47.

Plating of a Battle-ship.—The outer bottom plating of a battle-ship is made 25 lbs. ($\frac{5}{8}$ in.) over the length of the double bottom, and 20 lbs. ($\frac{1}{2}$ in.) at the ends. The fore-and-aft rows of plating, called *strakes*, are riveted to one another by double riveted laps. The plating is worked on the raised and sunken system as shown in Fig. 47. At every frame in way of each outside strake a filling piece or *liner* is necessary to get good riveting between the frame and the plate. These liners are often lightened, as shown, by oval holes between the rivets. At the middle line an additional outside plate, 30 lbs. ($\frac{3}{4}$-in.), is worked, called the *outer flat keel*, the inner plate being the *inner flat keel* (see Fig. 3).

Riveting in Outer Bottom.—Fig. 46 shows in some detail the arrangement of the riveting, etc., in the outer bottom plating near the middle line. The lower angles of the vertical keel are connected to the flat keels by 1-in. rivets, and as the vertical keel has to be watertight, these rivets on one side are closely spaced, viz. $4\frac{1}{2}$ to 5 diameters. On the other side a wider spacing, viz. 8 diameters, is all that is necessary. Close spacing is necessary for the watertight longitudinal, No. 2, and wide spacing for the non-watertight longitudinal, No. 1. The butt-straps of the inner and outer flat keel are treble riveted with 1-in. rivets, and extend from the keel angles to the edges of the plate concerned. The edge of the inner keel is connected to the outer keel by a single

row of rivets, to get a good connection and to allow the edge to be caulked. The edge riveting of the outer flat keel is double riveted with 1-in. rivets. The remainder of the riveting is $\frac{7}{8}$ in. diameter, as shown. The butt-straps are double riveted, and for the outside strakes extend from the longitudinals to the edge of the inside strakes. For the inside strake the strap extends the whole width of the plate.

The ordinary frames are connected to the outer bottom by $\frac{7}{8}$-in. rivets spaced 7 to 8 diameters. For the watertight frames it is necessary to have the rivets closely spaced, viz. $4\frac{1}{2}$ to 5 diameters. This close riveting cuts away a lot of material from the outer bottom plating in one transverse section of the ship, and causes this section to be a distinct line of weakness. In order to compensate for this, a wide liner called a *bulkhead liner* is fitted to each outside strake instead of the ordinary liner. This wide liner forms a sort of strap over the weak place, and in this way the strength at the watertight frame can be brought up to an equality with that at an ordinary frame. In the case of the second strake from the keel in Fig. 46, it will be noticed that the watertight frame comes next to a butt-strap, and the strap is made wide enough to act as a bulkhead liner and butt-strap combined.

Fig. 46A shows a form of bulkhead liner adopted in some ships to economize weight. The outside strake is only pierced at every other rivet, so that the reduction of strength is not so great at the section, and the liner can be smaller than would otherwise be necessary.

The side plating above the protective deck is recessed back from the side of the ship to make room for the armour and backing (see Fig. 18). This plating is in two thicknesses, each of 20 lbs. ($\frac{1}{2}$ in.), for 9-in. armour. No butt-straps or edge strips are necessary for this plating, as each thickness acts as security to the edges and butts of the other thickness. Above the armour the plating is 20 lbs. ($\frac{1}{2}$ in.) to the upper deck, except where an armoured battery is worked between the main and upper decks, as Fig. 13, in which case the double thickness of plating is carried right up.

The side plating is doubled in way of any protective plating to form a flush surface. Thicker or doubling plates are also worked in way of the stem and sternpost, in way of hawse-pipes, and where necessary to take the chafe of anchors and cables, and

in other places where special local strength is required, as in wake of torpedo-tubes. Covering plates are worked at the lower and upper edges of the side armour, as Figs. 18 and 34, connected to the armour by tap rivets. The edges of these plates are caulked so as to make the joint at the armour watertight.

In some battle-ships and cruisers the topside above the upper deck is worked to the boat deck, forming a closed-in superstructure (see Fig. 22). This plating is not intended to take any structural strains, and the sides are severed by the gangway ports. The

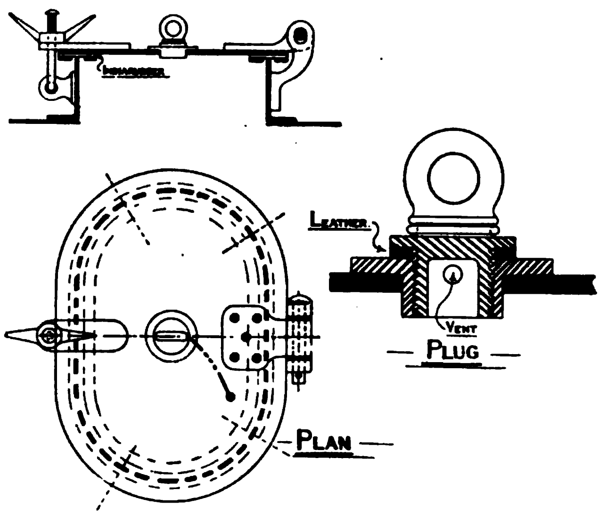


Fig. 48.—Watertight manhole.

plating of the boat deck is also severed, and a sliding joint is made as shown in Fig. 5.

Inner Bottom.—The inner bottom of a battle-ship is generally 15 lbs. ($\frac{3}{8}$ in.), with the middle line strake 20 lbs. ($\frac{1}{2}$ in.). This plating extends to the fourth longitudinal, and the inner skin is continued to the protective deck by means of the wing bulkhead, which is 15 lbs. It is most important that convenient access should be provided to all the watertight compartments of the double bottom, in order that the condition of the plating, etc., may be periodically examined. The best arrangement would be to have the two manholes at opposite corners of the compartment, so that the space might be well ventilated before entering. Owing, however, to the obstructions caused by the boiler bearers, etc., this

is not generally possible; an actual arrangement adopted in one ship is shown in Fig. 49. The manholes should be on raised coamings, with a hinged cover, secured by a number of butterfly nuts as shown in Fig. 48. In some cases a hinged cover cannot be obtained, owing to some obstruction, and in these cases the cover is connected to the coaming by a chain. A plug is fitted to the manhole cover to enable the compartment to be sounded when necessary. This plug is hollow, as shown in Fig. 48, with a small vent-hole at the side. When the plug is slightly unscrewed, this hole allows the air to escape when flooding, and will also show when the compartment is full. The plug can then readily be screwed up tight. It is very necessary that the air should be allowed to escape so that the tank may be completely filled. At

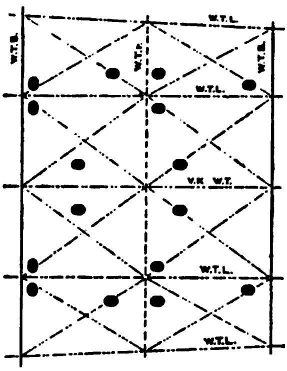


FIG. 49.—Manholes in inner bottom.

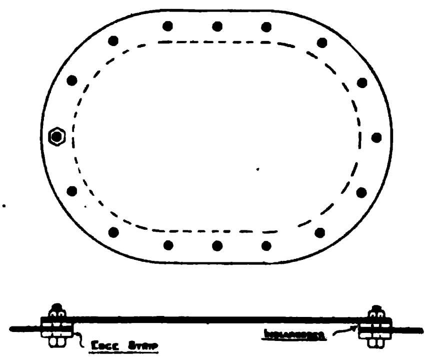


FIG. 50.—Watertight manhole on bulkhead.

other places, as on watertight flats or bulkheads, where occasional access through is necessary for inspection, the edge of the hole is stiffened by a strip, and a portable plate is secured by studs as shown in Fig. 50.

We have already noticed that although the inner bottom proper ends about one-sixth the length from each end, yet a virtual inner bottom is obtained well to the ends by the watertight flats, etc.

These double bottom spaces are carefully tested for watertightness by filling with water, the pipe conveying the water giving a head of about 5 ft. above the L.W.L. If the test is not satisfactory, the defects have to be made good and the compartment retested.

Outer and Inner Bottom Plating of a First Class Cruiser.

—This plating is arranged on the same principles as for a battle-ship, the thickness generally being somewhat less. Special attention, however, is paid to the structure at the keel and at the upper deck. Fig. 3 shows the structure at the keel of a cruiser whose ratio of length to depth is 12·4. This is shown in comparison with the structure at the keel of a battle-ship of greater displacement whose ratio of length to depth is only 9·4. It will be noticed that the

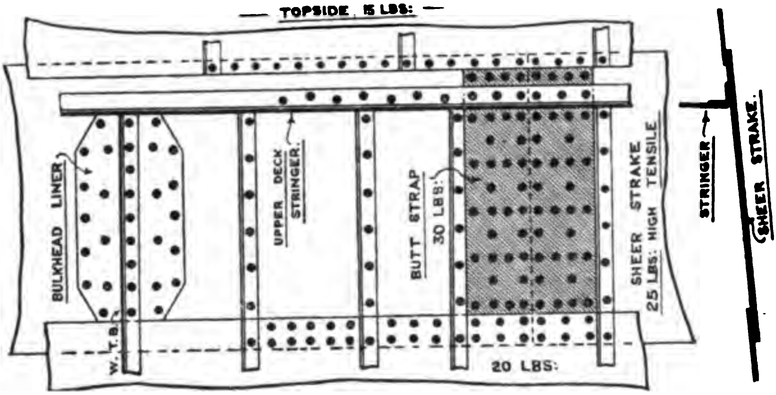


FIG. 51.—Sheer strake of cruiser.

vertical keel is deeper, and the outer flat keel is 45 lbs. (nearly $1\frac{1}{8}$ in.), as against 30 lbs. ($\frac{3}{4}$ in.) in the battle-ship. The middle strake of the inner bottom is 25 lbs. as against 20 lbs. in the battle-ship. Fig. 51 shows the sheer strake of this cruiser, *i.e.* the plate of side next the upper deck. It is of 25-lb. high tensile steel, and the butt-strap is 30 lbs., quadruple riveted.

The inner bottom plating of a large cruiser presents no special features, and the remarks made above apply in this case also. It has already been noticed that the wing bulkhead has been dispensed with in recent ships to give a greater coal capacity. Figs. 52 and 53 show the whole of the watertight subdivision of a large cruiser, in which it will be noticed that an inner skin is obtained well towards the ends by means of the flats, etc., to the magazines and store-rooms.

Plating of a Second Class Cruiser.—The outer bottom plating of the cruiser shown in Fig. 25 is generally $17\frac{1}{2}$ lbs. ($\frac{7}{16}$ in.). The flat keel is 25 lbs. ($\frac{5}{8}$ in.), and the sheer strake 25 lbs. The inner bottom is $12\frac{1}{2}$ lbs. ($\frac{5}{16}$ in.).

Plating of a Third Class Cruiser.—The bottom plating of the

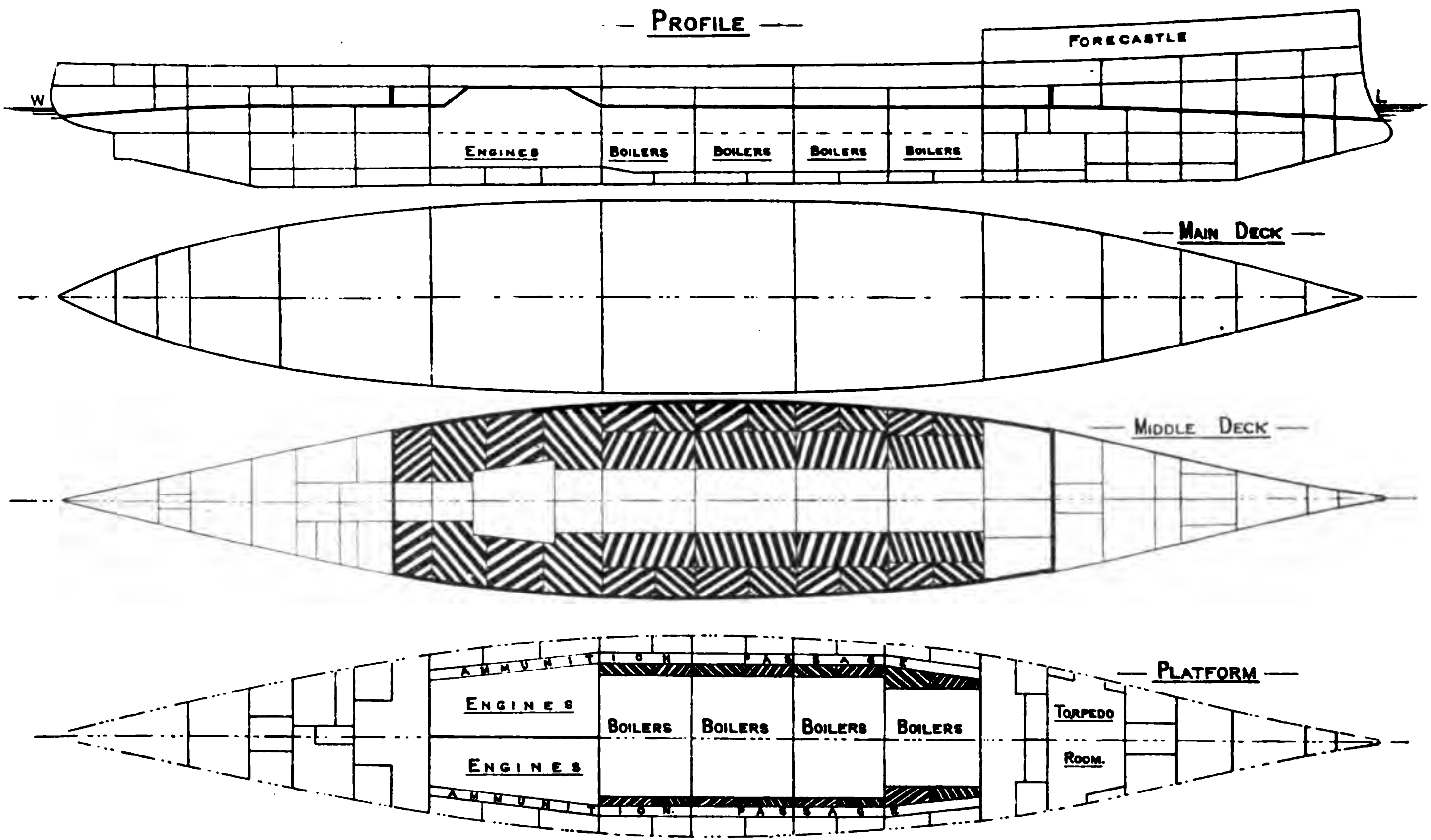


FIG. 52.—Watertight subdivision, first class cruiser.

cruiser shown in Fig. 28 is generally 12½ lbs. ($\frac{5}{16}$ in.). In way of the engine-room, however, it is 17½ lbs. ($\frac{7}{16}$ in.), to stiffen the ship in way of the fast-running machinery. The flat keel and sheer strake, next the upper deck, are both 20 lbs. (½ in.). These are both reduced to 17½ lbs. beyond the half length.

Plating of Sloop.—The bottom plating of the sloop shown in Fig. 30 is generally of 10 lbs. (¼ in.), with flat keel and sheer strake of 15 lbs. (⅜ in.).

Plating of Destroyer.—For this type of vessel high tensile steel is used for the outer bottom plating in the more recent ships. The flat keel in Fig. 32 is 36 in. × 10 lbs. (¼ in.), sheer strake 30 in. × 8½ lbs., the remainder of the plating being 6½ lbs.

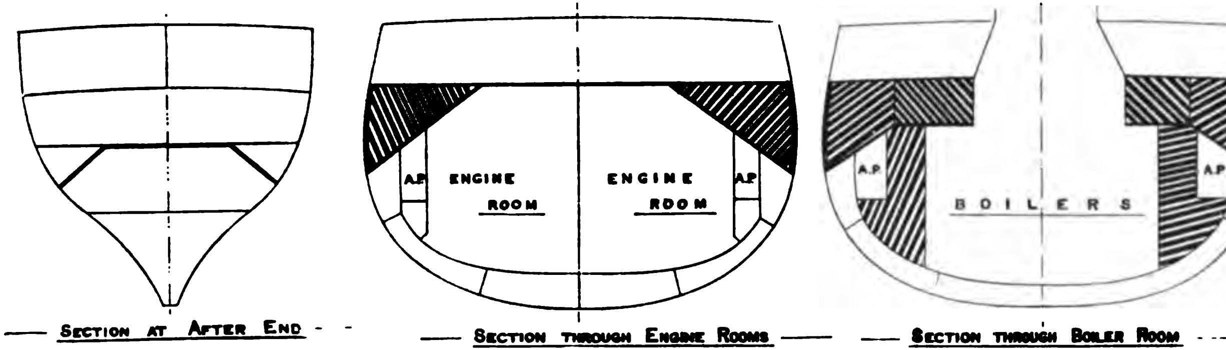


Fig. 53.—Watertight subdivision, first-class cruiser.

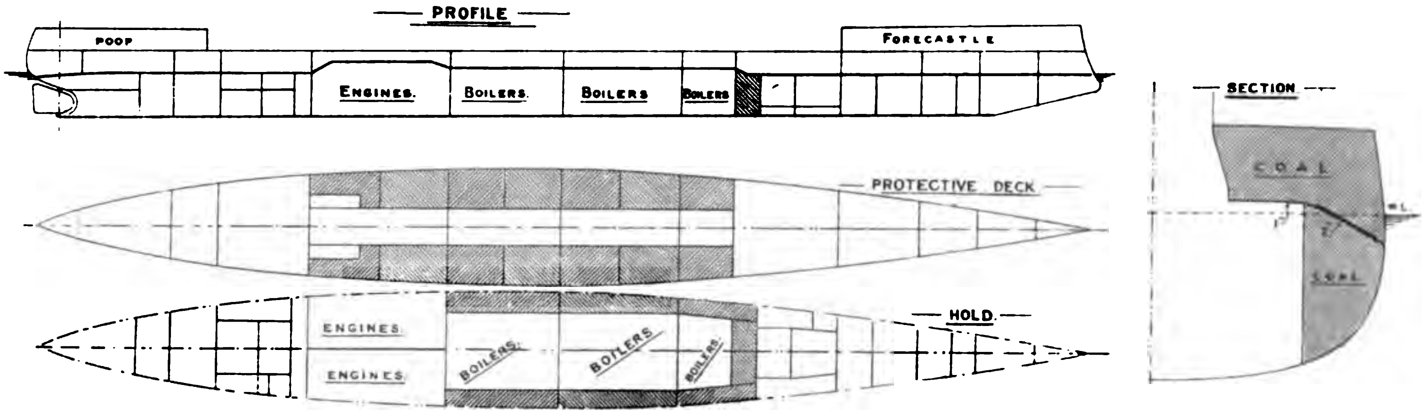


Fig. 54.—Watertight subdivision, third class cruiser.

CHAPTER VI.

WATERTIGHT BULKHEADS, DOORS, ETC.

THERE are four main methods of watertight subdivision, viz. by means of

(i.) A watertight inner bottom with watertight vertical keel, longitudinals, and frames;

(ii.) Watertight decks and flats;

(iii.) Transverse bulkheads; and

(iv.) Longitudinal bulkheads.

We have already dealt with the first two of these. The valuable feature of a double bottom has to be dispensed with in small vessels on account of the space thus occupied. In all ships, however, we get watertight subdivision from the last three of the above (see Figs. 52 and 54).

We now deal with the bulkheads. These are not only useful, in dividing the ship into a number of watertight compartments, but they form a most valuable addition to the ship's structural strength.

Transverse Bulkheads.—These are watertight partitions which go transversely across the ship. Fig. 52 shows the large number of such bulkheads fitted in a large cruiser; Fig. 54 is for a small cruiser. The one nearest the stem, extending to the upper deck, is the *collision bulkhead*, and many instances have occurred, especially in merchant vessels, in which, after collision, this bulkhead has remained intact and saved the ship from possible foundering. On account of its importance it is well stiffened, and in recent ships no openings of any kind are allowed in it. Any access required to the forward side must be by means of scuttles through the decks, and if the forward space does require draining, it must be pumped out by means of a hose.

In some recent battle-ships an additional bulkhead is fitted 3 ft. abaft the collision bulkhead. This is termed the "cofferdam"

bulkhead, and the 3 ft. space thus formed is intended to be packed, like an ordinary cofferdam, before ramming, to limit the flow of water aft, supposing the collision bulkhead to be damaged (see Fig. 67).

A governing feature in the construction of any bulkhead is the area and depth of unsupported plating likely to be exposed to water pressure. The transverse bulkheads forward and aft of the machinery spaces are well supported by the decks and flats, and so do not require any extensive stiffening, as the unsupported area is not great. Such bulkheads are usually formed of 10-lb. ($\frac{1}{4}$ in.) plating, stiffened with angles $3\frac{1}{2}$ in. $\times$ $2\frac{1}{2}$ in., worked vertically, the spacing varying from 2 to $2\frac{1}{2}$ ft. The plating is lap jointed at edges and butts, and single riveted. Bulkheads forming the sides of magazines in which teak lining is fitted are stiffened by

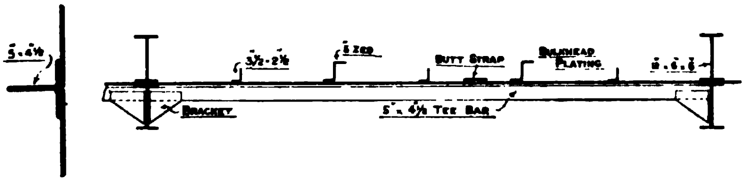


Fig. 55.—Stiffening to bulkhead.

zed bars 3 in. deep. In the most recent ships, however, this is not necessary, because of the omission of the lining to magazines.

The collision bulkhead is formed of 15-lb. ($\frac{3}{8}$ in.) plating, stiffened by 5-in. zeds and $3\frac{1}{2}$ in. $\times$ $2\frac{1}{2}$ in. angles spaced alternately 2 ft. apart. Above the protective deck the angles only are fitted.

The transverse bulkheads forming the divisions between the engine and boiler-rooms are specially constructed and stiffened because of the very large area and depth of unsupported plating. (In one case 46 ft. wide and 25 ft. deep.) The plating is 15 lb. ($\frac{3}{8}$ in.), worked flush jointed, the horizontal joints being covered with a tee bar $4\frac{1}{2}$ in. $\times$ 5 in. forming the edge strip. The vertical joints are covered with single-riveted butt-straps on the opposite side to the tee bars. The main stiffening is worked vertical, and is formed of 5-in. zeds every 4 ft., with $3\frac{1}{2}$ in. $\times$ $2\frac{1}{2}$ in. angles between. At intervals of about 8 or 12 ft. the zeds are replaced by I bars 12 in. $\times$ 6 in. $\times$ 6 in., worked on *both sides* of the bulkhead, the tee bars being cut and connected to them (Fig. 55).

Both the zed and I bars are well supported at the head and heel by bracket plates. This extensive stiffening has been found necessary to enable the bulkheads to withstand the great pressure that would exist supposing one of the adjacent compartments filled with water. It is the practice in each ship while building to actually fill a boiler-room with water to a height of 5 ft. above the load waterline, in order to test the strength of the bulkhead. The safety or control of a ship might very conceivably depend on one of these main bulkheads remaining intact if an engine-room or boiler-room were flooded.

The bulkheads forming the fore end of the fore boiler-room and the after end of the engine-room do not need this extensive stiffening, because of the support received from the decks and platforms. In these cases the 12-in. I stiffeners are not fitted.

The transverse bulkheads in way of the inner bottom are bounded thereby, and the watertightness is continued to the outer bottom by means of the watertight frames already considered.

Between the main transverse bulkheads, divisional bulkheads are fitted in the side upper and lower bunkers, as seen in Fig. 52. Beneath the watertight flat in the upper bunkers of battle-ships, shown in Figs. 12 and 13, a further set of bulkheads is fitted between the above, thus giving most minute subdivision to the side in the neighbourhood of the waterline.

The fore-and-aft longitudinals, vertical keel, etc., are worked continuously through all the transverse bulkheads in order to maintain continuity of the longitudinal strength. Where the longitudinal takes the form of a zed and angle, as at the ends of a ship, the watertightness is secured by working angle collars round, as shown in Fig. 56.

FIG. 56.

It will be noticed in Figs. 52 and 54 that a number of the bulkheads are carried right up to the upper deck. This is important in view of the sinkage, heel, and change of trim that might ensue after damage. The bulkheads being carried well above water, there is more likelihood of confining the water on one side.

Longitudinal Bulkheads.—There are a number of small longitudinal bulkheads forming the boundaries of magazines, etc. These assist in maintaining the watertight subdivision, but being

only of small area are of 10 lb. ($\frac{1}{4}$ in.) only, with $3\frac{1}{2}$ in. × $2\frac{1}{2}$ in. angle stiffeners. For the sides of magazines with teak lining 3-in. zeds were formerly used as the stiffeners.

The principal fore and aft bulkheads in a large ship are—

(i.) Middle line engine-room bulkhead.

(ii.) Inner coal-bunker bulkhead.

(iii.) Outer coal-bunker or wing bulkhead.

(iv.) Upper coal-bunker bulkhead.

These are all shown in Figs. 52 and 53. We have noticed that in some cases (Fig. 23) the two latter bulkheads have been omitted to increase the coal capacity and to make the transport of coal more easy.

(i.) *Engine-room bulkhead.*—This bulkhead extends for the whole length of the engine-room, and is taken to the height of the main deck. In battle-ships this is only necessary in the ventilators, etc.; but in large cruisers, where the protective deck has to be lifted to the main deck (Fig. 52), the bulkhead is carried right up to this deck. The bulkhead is carried to this height in order to prevent water flowing over the top in case one engine-room was flooded. It is important that this bulkhead should be amply strong in view of the severe strains that would come upon it if one engine-room were flooded. On this account it is of $12\frac{1}{2}$ lb. ($\frac{5}{16}$ in.), and constructed and stiffened in a similar manner to the main transverse bulkheads.

(ii.) *Inner coal-bunker bulkhead.*—This bulkhead is well supported by the divisional bulkheads in the coal bunkers, and is not so strongly stiffened as in the previous case. The plating is lap-butted, lap-jointed, single-riveted; the two lower strakes are $12\frac{1}{2}$ lb. ($\frac{5}{16}$ in.), and the remainder 10 lb. ($\frac{1}{4}$ in.). The stiffeners are 5-in. zeds at each beam, connected above and to the inner bottom by bracket plates. Angles $3\frac{1}{2}$ in. × $2\frac{1}{2}$ in. are worked between.

(iii.) *Wing bulkhead.*—This bulkhead when fitted forms a virtual continuation of the inner bottom, and is formed of 15-lb. ($\frac{3}{8}$ in.) plating, stiffened the same as the inner bulkhead.

(iv.) *Upper coal-bunker bulkhead.*—This bulkhead, only being from deck to deck and not likely to have to stand any great pressure of water, is of 10-lb. ($\frac{1}{4}$ in.) plating, stiffened by angles $3\frac{1}{2}$ in. × $2\frac{1}{2}$ in. worked 2 ft. apart.

The beams of the ship run continuously through the longitudinal bulkheads, and in order to make the upper portion

watertight, angle bars are smithed to fit in the space between the beams, as Fig. 57. The riveting is closely spaced and the whole carefully caulked.

The bulkheads underneath barbettes are specially stiffened to efficiently support the barbette, etc.

Numbering of Bulkheads.—Transverse bulkheads are distinguished by the number of the frame station at which they come. Longitudinal bulkheads are distinguished by the numbers of the frames at which each begins and ends. Thus, bulkhead 116 to 140 would be the bulkhead extending between frames 116 and 140. Formerly bulkheads were distinguished by letters, but it has been found more convenient to use numbering instead.

Water-testing.—All watertight flats and bulkheads, outer bottom, inner bottom, and decks are tested for watertightness under water pressure. Wherever practicable, the compartments

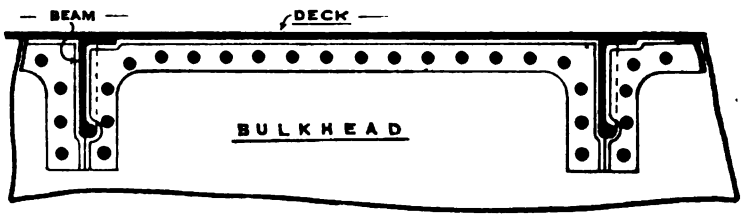


FIG. 57.

are actually filled with water, the *head* of water applied being about 5 ft. above the load waterline. Where this is not possible, the watertightness is tested by means of a hose. Usually one of the boiler-rooms and one of the engine-rooms are filled with water to a height of 5 ft. above the load waterline, in order to test the *strength* of the bulkheads, as well as the watertightness.

A large number of holes are necessary in a bulkhead for the purpose of connecting fittings, etc., to the bulkhead. It is important to note that if any such fittings are removed for any purpose, the holes left behind should be properly filled up again by a tap rivet. Otherwise the watertightness of the bulkhead is destroyed. The glands on bulkheads, where voice-pipes, electric wires, etc., pass through, should be periodically examined to see that the screws for attachment have not worked out.

An Admiralty circular has recently been issued, directing that

one compartment is to be flooded in each ship at least once a year, to test the watertightness of the bulkheads, doors, etc. (see S. 32111/1903, January 29, 1904).

Watertight Doors.—The transverse and longitudinal bulkheads being watertight, it is necessary that openings in them should be capable of being made watertight. These openings are made as few in number as possible, but still a number of

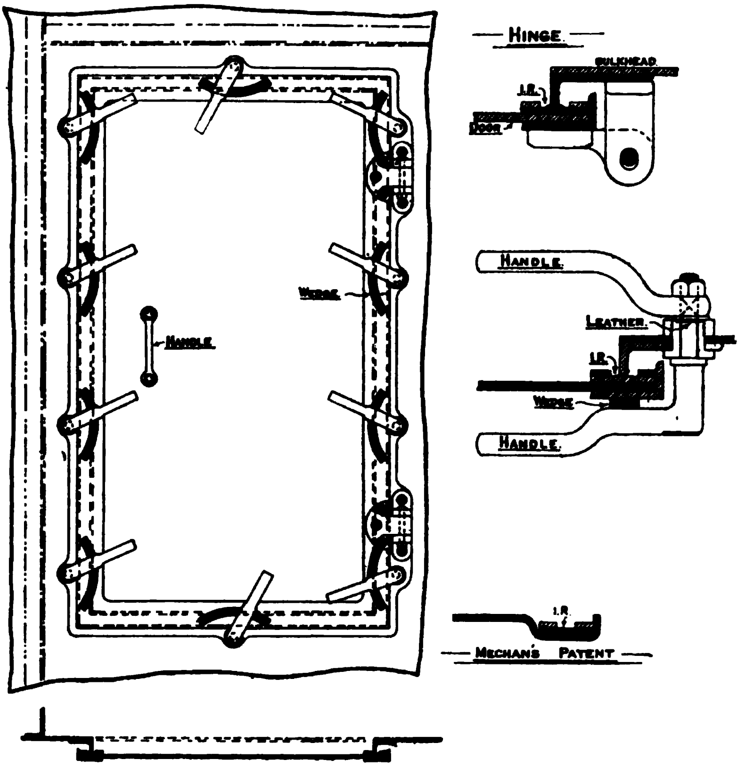


FIG. 58.—Hinged watertight door.

the most important bulkheads have to be pierced by doors, which must be open for passage through, even supposing the ship to be in action. Such bulkheads are—

(*a*) The divisional bulkheads between the engine and boiler-rooms.

(*b*) The coal-bunker bulkheads.

(*c*) The bulkheads at the ends of ammunition passages.

Watertight doors are of three kinds, viz. hinged, vertical sliding, and horizontal sliding.

Hinged Doors.—This type of door is the most common in H.M. ships, being fitted to the less important bulkheads below water, and to nearly all the watertight bulkheads above water. (In a few cases above water, a hinged door would be inconvenient, and a sliding door is fitted, as in the upper-coal bunkers.) No provision is made for closing these hinged doors from above, they must be closed at the door itself.

The opening made in the bulkhead is stiffened round by an angle bar frame (Fig. 58). The door is formed by a piece of plate of the same thickness as the bulkhead; this also has a stiffening bar round the edge. In "Mechan's" patent door this stiffening is obtained by embossing the edge to the shape shown, giving us a lighter door. India-rubber is worked all round the edge of the door as shown, which engages with the stiffening bar on the bulkhead. The door is forced home to the rubber by means of handles, each of which presses against a wedge-shaped piece of metal on the door. Spring clips are placed to keep the handles away from the door when not in use. The hinges are made with an elongated hole for the pin, so that a certain amount of play is allowed to force the door home. The handles pass through a metal collar on the bulkhead, and watertightness is obtained by a leather washer. The india-rubber on the door, when perished, can be readily removed by means of the securing strips, which screw into the door. This india-rubber must never be painted or greased. It should be chalked to prevent sticking. The leather washers on the door clips should be periodically examined. No tread over the sill should be allowed, which will prevent the door from closing.

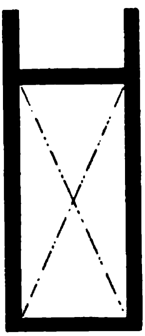


Fig. 59.

Vertical Sliding Doors.—These doors are fitted to bulkheads where it is necessary to close from above, and where sufficient head room is possible above the opening to take the door when open. They are of two kinds—(1) ordinary, and (2) quick closing. These latter are fitted on the bulkheads between the engine and boiler-rooms. Quick-closing doors were formerly fitted with balance weights, to render the closing easy and quick. Now, however, a coarse thread of large diameter is used on the spindle. For ordinary doors a

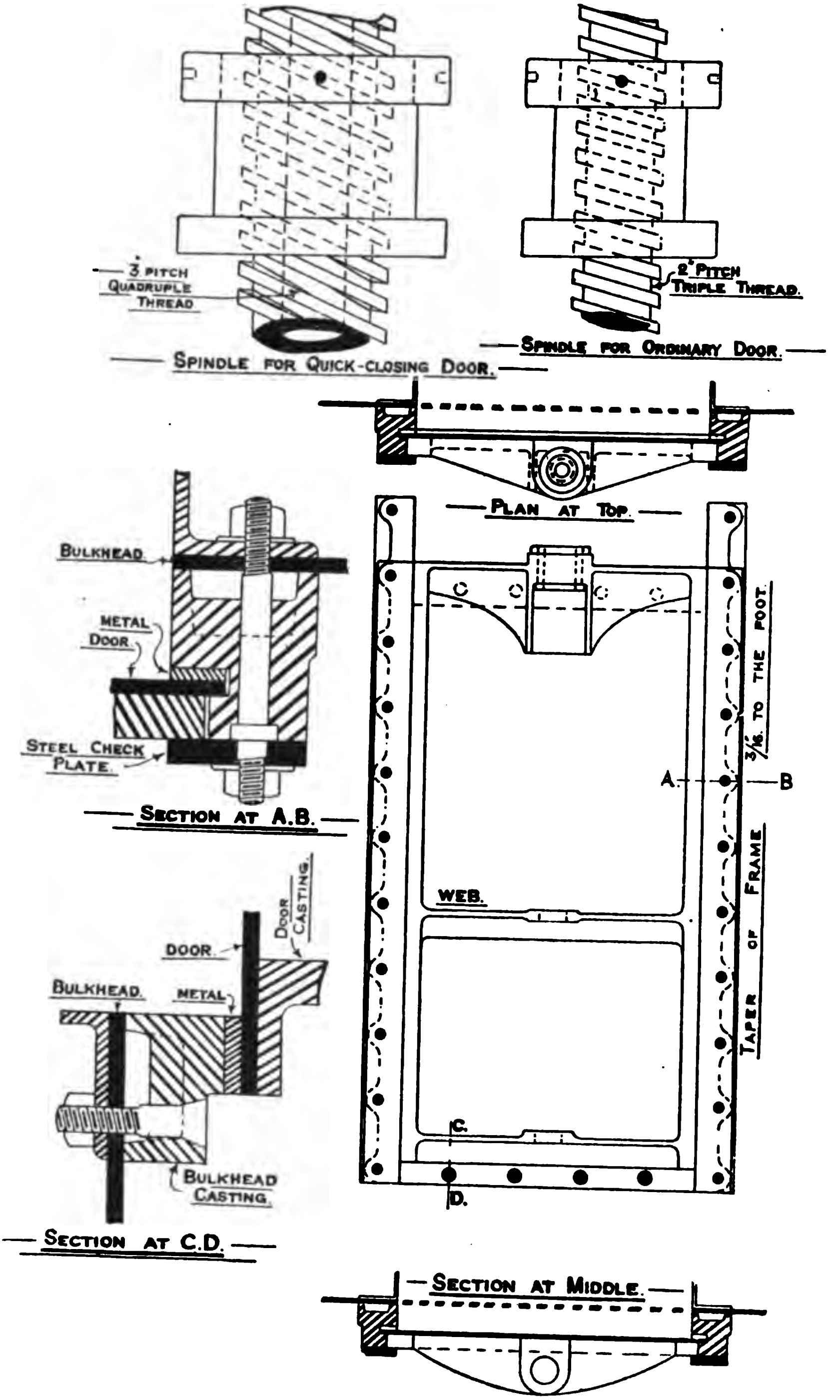


Fig. 60.—Vertical sliding watertight door.

smaller thread of less pitch is used. Otherwise the construction of the doors is substantially the same (Fig. 60).

A cast-steel frame, shaped as shown in Figs. 59 and 60, is bolted to the bulkhead. The sides are formed, with a taper of $\frac{3}{16}$ in. to a foot, to correspond with the frame on the door. The door is formed of a steel plate, with a steel casting to stiffen it, shaped as shown. The section of the door shows very clearly the construction, and how the door jams in between the bulkhead casting and the check plate. The top edge of the casting is provided with a metal nut, with a thread to correspond with that on the spindle. This spindle extends from the top of the opening to the *main deck*, the thread, of course, extending only sufficient to open the door. The door can be closed by a set of gearing from below, as well as from above, and, when necessary for purposes of escape, it can be worked from either side of the bulkhead.

The doors between engine and boiler-rooms, and to coal-bunkers, have their sills kept well above the inner bottom.

Horizontal Sliding Doors.—Doors of this type (Fig. 62) are fitted to important bulkheads below water (and to a few above water), to which vertical doors cannot be fitted because of the lack of sufficient head room. The opening in the bulkhead is bounded by a steel casting, shaped as Fig. 61. This casting is secured to the bulkheads by bolts. At the bottom and top there is a groove, having a taper of $\frac{3}{16}$ in. to a foot, in which the door slides, and in which the door jams when closed. The door is formed by a steel plate, well stiffened by a steel casting, shaped as shown. The weight of the door is taken by two wheels. The inside edge of the door has a gunmetal strip all round, which engages with the bulkhead casting. The leading edge of the door jams into a number of clips on the edge of the frame, and the following edge of the door has some back pieces, which jam against the back of the bulkhead frame. The door has two horizontal steel racks, and into these two gunmetal pinions engage, these being rotated by a vertical spindle. This spindle extends to the *main deck*, so that the door can be worked from above if desired.

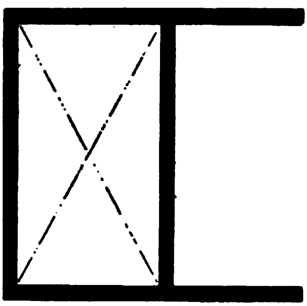


FIG. 61.

The door can also be closed at the door by a spanner, working on a nut on the spindle.

A disadvantage with horizontal doors is the existence of the bottom groove. The dirt from the traffic soon fills up the groove,

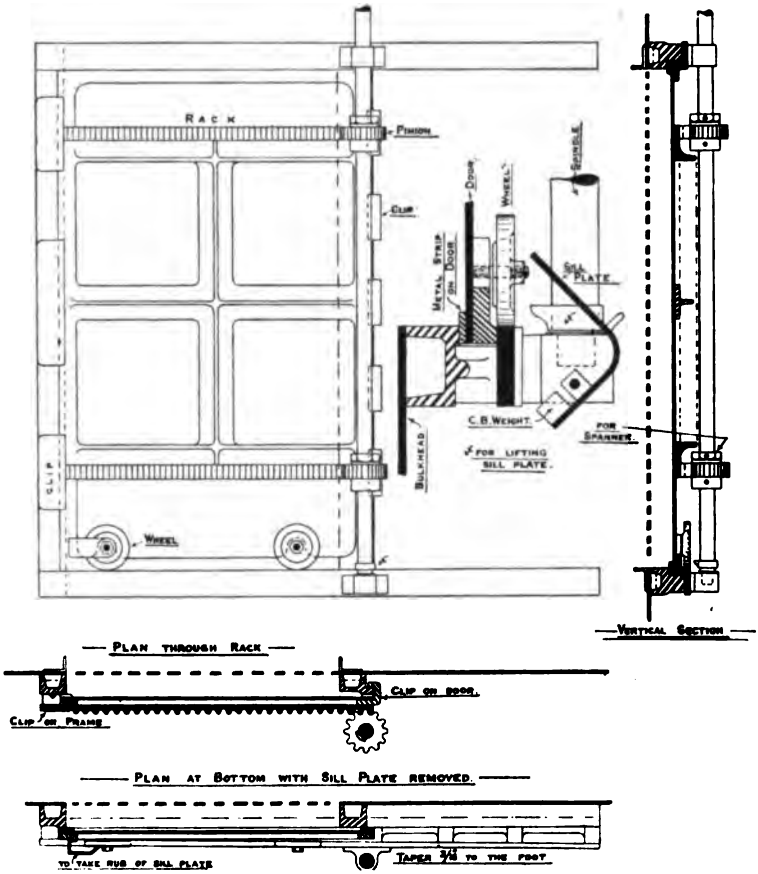


FIG. 62.—Horizontal sliding watertight door.

and this gets hard and prevents the door from closing. To obviate this a *sill plate* is fitted over the sill. When the spindle for closing is first turned, it turns the sill plate up out of the way, to allow the door to slide along and shut. When the door is again opened, the sill plate automatically drops over the groove again.

The vertical spindles for closing watertight doors are always so arranged that a right-hand motion closes the door. The deck

plate at the main deck, where the spanner is worked, is provided with an indicator to show whether the door is shut or not. The spanner, however, should always be turned as far as possible to close the door, as frequently after the gearing is worn the indicator may point to "shut" when the door itself is not completely shut.

Screens are fitted to the doors of coal bunkers on the inside to protect the doors from the pressure of the coal, and thus enable the doors to be opened and closed when the bunkers are full (see Fig. 120).

A book is supplied to the commanding officer of each ship giving a list of the watertight compartments, with their capacity in cubic feet, the boundaries and means of closing, and the places at which the doors can be closed. In addition to this the means of pumping and ventilating each compartment is given. A specimen page of such a book is given at the end of Chapter IX.

In the Admiralty circular above mentioned attention is drawn to the defects found in watertight doors, and the methods to be adopted to keep them efficient.

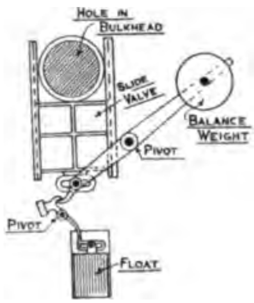
Sluice Valves are fitted to the lower parts of certain bulkheads to drain water from one compartment to another. These valves are worked like a small vertical door, with a spindle extending to the main deck, having an indicator on the deck plate. In recent ships no sluice valve is fitted on the collision bulkhead.

Automatic Ventilation Valves.—In the more recent ships the artificial ventilation is arranged so that the main transverse bulkheads are not pierced. In previous ships, however, the ventilation was provided by a series of fore-and-aft trunks, having louvres at intervals (see Fig. 105). These trunks were supplied with air from large steam-driven fans. In this way the watertightness of the bulkheads was completely destroyed, and the openings had to be provided with *automatic valves*, arranged to shut automatically if water rose in the compartment on either side. Provision was also made for shutting all these valves if desired by pipes led from the main or upper deck.

Two of these valves are shown in Figs. 63 and 64. Both operate by a float, which rises if water enters, and this float then releases a balance weight, which closes the opening. In the case of Beck's valve (Fig. 63) an ordinary slide valve is used. In Broadfoot's valve (Fig. 64) the valve turns round and closes the

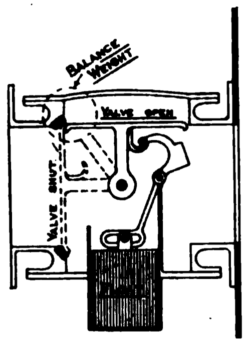
opening. In either case the small opening in the bulkhead for conveying water to the float from the opposite side is automatically closed when the balance weight falls.

These valves under the conditions on board ship, are found to



—Beck's Automatic Valve—

Fig. 63.



—Broadfoot Automatic Valve—

Fig. 64.

be far from efficient, and require jamming home by hand. All the bearing surfaces should be kept clean (see Admiralty circulars mentioned below).

Inspection of Watertight Doors.—As so many doors have to be left open for access throughout the ship, even supposing an action is proceeding, it is obviously of the highest importance that they should all be in perfect working order, and the crew well exercised in closing them, so that they can be readily closed in any time of emergency. The following extracts from the "Steam Manual" gives the instructions regarding this:—

"The watertight doors and sluice valves are placed under the charge of the Chief Engineer. All the Engineer Officers must make themselves acquainted with the positions and methods of closing of watertight doors. All watertight doors are to be kept clear for immediate closing. No fitting of any kind is to be allowed which will require to be removed before the door is closed. Watertight doors and sluice valves are to be opened and shut regularly *once a week* to ensure their being in good working order. The Chief Engineer, besides preparing the station bill, is to take such measures as he may deem necessary to ensure that every person under his control shall know his post and be capable of performing his duty, so that in case of emergency the watertight doors and sluices may be closed without confusion."

Similar instructions are given in the King's Regulations.

Recent instructions have been issued in Admiralty circular, S. 32111/1903, of January 29, 1904, in regard to watertight doors and hatches. See also S. 31157/1903, of January 9, 1904, for Admiralty circular referring to H.M.S. *Prince George.*

Automatic Doors.—The following extract from Sir William White's report * on the loss of the *Victoria* may be quoted with reference to the suggestion frequently made that automatic or "self-closing" doors should be adopted instead of existing arrangements:—

"This suggestion is a revival of one made long ago, then carefully considered and put aside after certain experimental doors had been tried.

"Automatic arrangements are applied in valves to ventilating trunks and other small openings in bulkheads and platforms. Even in such cases the feeling of the Naval Service has led to the automatic fittings being supplemented by the means of closing the valves when desired. In doors and scuttles the risks of the automatic appliances failing to act, or of solid materials being carried into openings by a rush of water, and preventing doors from closing properly, would be much greater. These considerations have led to the retention of existing fittings, the design of which provides that, when properly closed and secured, doors and hatchway covers shall be as strong as the neighbouring partitions, and watertight under considerable pressure.

"There is no difficulty in making automatic appliances. It is a question of what plan secures the maximum of safety under the working conditions of the Royal Navy. With large numbers of disciplined men, familiar with the fittings, and constantly drilled in their use, it is possible to close and properly secure all the doors, etc., in a battle-ship in three to four minutes, or possibly a less time for ships after long periods in commission.

"In the *Victoria*, no orders were given to close doors until one minute before collision. It is established by the evidence that the doors, etc., were in good order. The failure to close doors, therefore, was due entirely to the insufficiency of time available, especially in compartments breached by the collision.

"Under these circumstances no new argument in favour of the use of automatic doors seems to arise out of the loss of the *Victoria*."

* Parliamentary Paper, No. C. 7208/1893.

CHAPTER VII.

STEMS, STERNPOSTS, RUDDERS, AND SHAFT BRACKETS.

Stems.—The simplest form of stem is that formed by a flat bar, to which the plating at the forward end is secured. This form of stem is adopted in merchant vessels and in the smaller classes of ships in the Royal Navy, as third class cruisers and destroyers. For larger war vessels, however, a stronger form of stem is necessary, because it is desirable that such vessels should be able to effectively ram an enemy's ship, without at the same time sustaining serious damage herself. The most effective form of stem

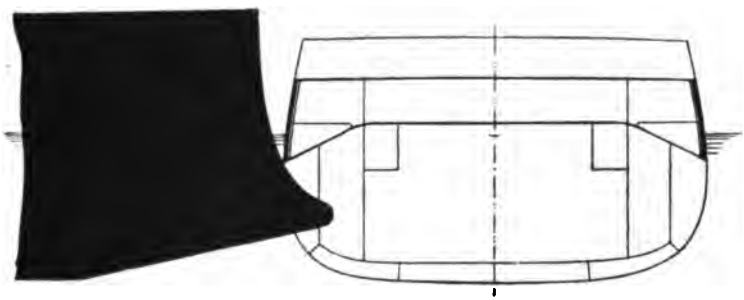


FIG. 65.

for this purpose is one having a ram below water, projecting well forward, so that it shall damage the slight under-water portions of the structure well in from the side before being brought up by the strong structure of the armour or protective deck of the other vessel (see Fig. 65).

Stems of steel vessels are now made of cast steel, a material possessing good strength and ductility (see Chapter II.), and capable of being cast into most efficient forms for the special purpose required. These castings are a great advance on the iron forgings

formerly in use, and a much more efficient ram has been by this means rendered possible.

When a vessel is sheathed with wood and copper we cannot use cast steel for the stem because of the galvanic action that would in all probability ensue between the copper and the steel. In such ships, therefore, the copper alloy, *phosphor bronze*, is used, but on account of the low strength of this material, the

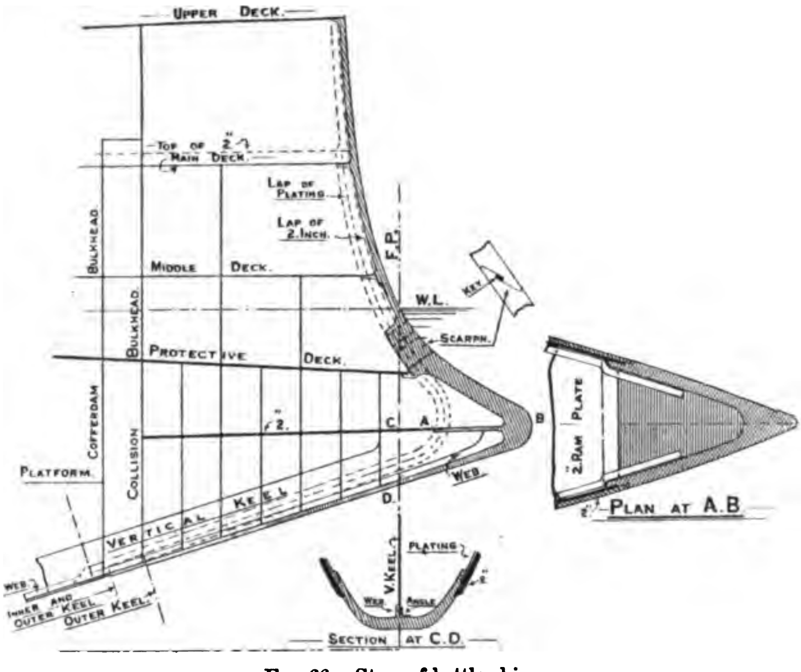


FIG. 66.—Stem of battle-ship.

casting has to be much more massive than a corresponding one of cast steel.

In the earlier ships with cast steel stems (*Royal Sovereign* to *Canopus*), the stem casting was carried well down into the body of the ship (Fig. 66), and on this account it had to be made in two pieces, because of the difficulties attending the manufacture and transport of such a large and intricate casting. The two pieces were connected together, as shown, by a scarph (Fig. 66), a tapered key being driven in to draw the parts together, and the whole well secured by screw bolts. The scarph was necessarily

a place of weakness, and this has been avoided in more recent ships by making the casting all in one piece. In order to do this the lower portion is made much shorter (see Fig. 67). This sketch shows the general shape of the casting, the ship being swelled out in way of the ram. Inside the ram projection two webs are cast, one vertical and one horizontal.

It is most important to support the stem effectively by the

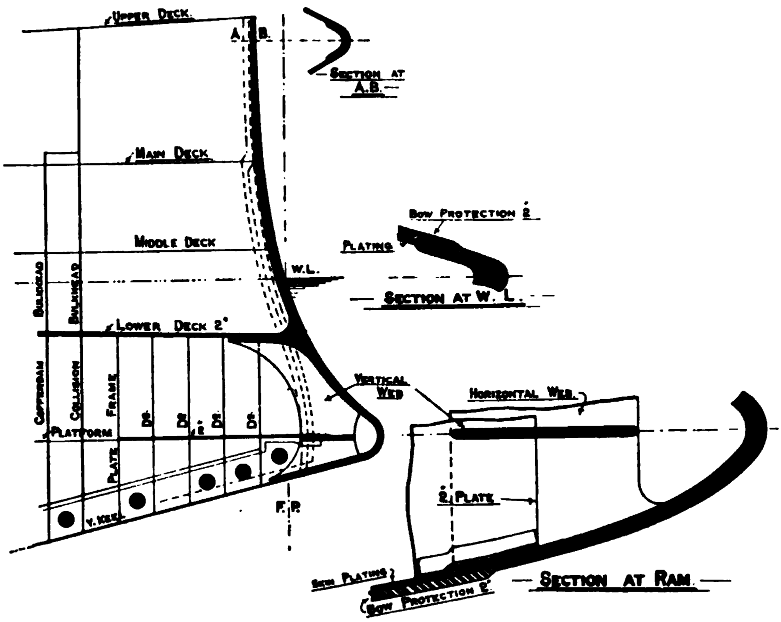


Fig. 67.—Stem of battle-ship.

adjacent structure. The following arrangements are made with this object in view, viz.—

(i.) The outer bottom plating is doubled in thickness and recessed into the casting.

(ii.) The bow protection, in this case 2 in., is recessed into the casting for one half its thickness.

(iii.) The lower deck, which is a thick deck, is well connected to a large projection on the stem casting.

(iv.) At the level of the platform a 2-in. plate is worked, well connected to the horizontal web inside the ram. This 2-in. plate extends back to within 3 ft. of the collision bulkhead.

(v.) The vertical keel runs up to the stem, and is well connected to the vertical web inside the ram.

It is thus seen that every precaution is taken to support the stem to make it efficient for ramming purposes, and (iii.) and (iv.)

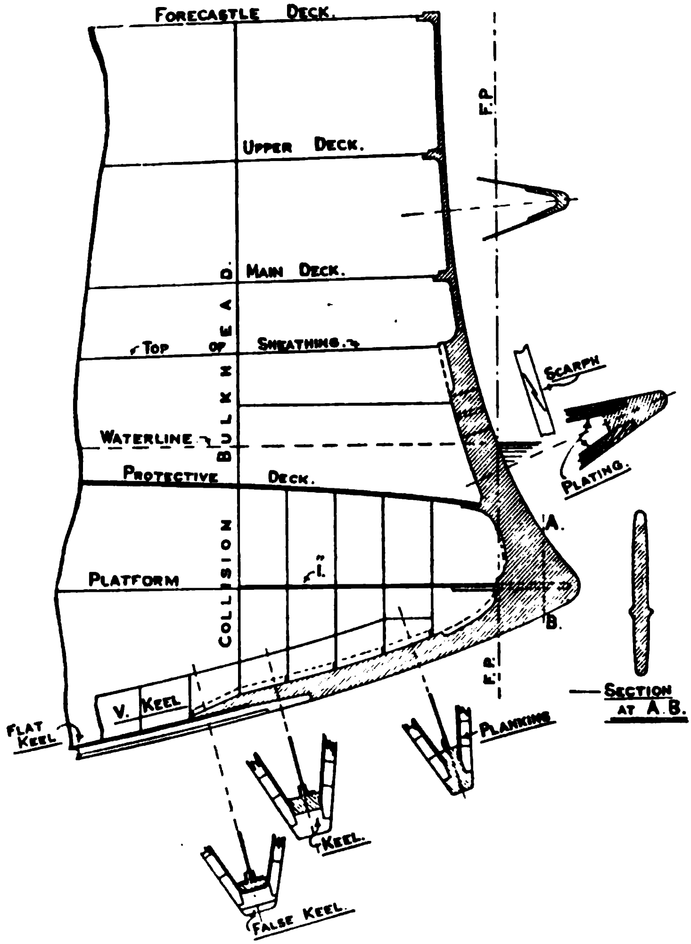


Fig. 68.—Stem of sheathed cruiser.

above not only provide a direct support, but they would resist the side bending action that would ensue when the ships swung together after the blow was struck.

The sketches each show a "cofferdam" bulkhead 3 ft. abaft

the collision bulkhead. It would be advisable to pack the space between these bulkheads, like an ordinary cofferdam, before ramming, to limit the flow of water aft, in case the collision bulkhead was damaged; access is obtained through the various decks for this purpose.

The phosphor bronze stem of a sheathed second class cruiser is shown in Fig. 68. In this case the stem is cast in two pieces, and the plating and planking have both to be recessed into the casting. The wood keel also has to be recessed as shown.

In the stems of recent ships the casting has been stopped at the main deck, the strength to the upper deck or forecastle being provided for by a bent plate.

Sternposts.—The remarks already made as to the necessity of phosphor bronze stems in sheathed ships apply in this case also.

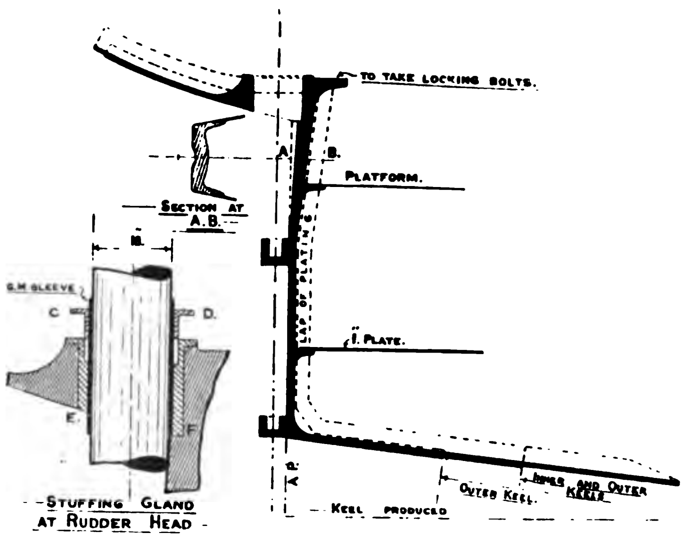


FIG. 69.—Sternpost of battle-ship.

The sternpost of a single screw ship has to be formed to receive the propeller as well as to form a support on which to hang the rudder. Very nearly all the vessels now in the Royal Navy are, however, twin screw, so that the main function of the sternpost is to receive the rudder.

The shape of the sterns of ships varies considerably in different classes, but in all war-ships (except the smallest) an essential

condition to be fulfilled is, that the stern shall be so formed that the rudder and steering gear are well below water and under protection. In order to do this the stern is carried well abaft the rudder-head, as seen in Figs. 71 to 76, to house the rudder-head and steering gear.

Taking first the case of battle-ships, the stern in vessels up to the *Canopus* class was of simple construction. The sternpost was a casting shaped as shown in Fig. 69, with projections on which

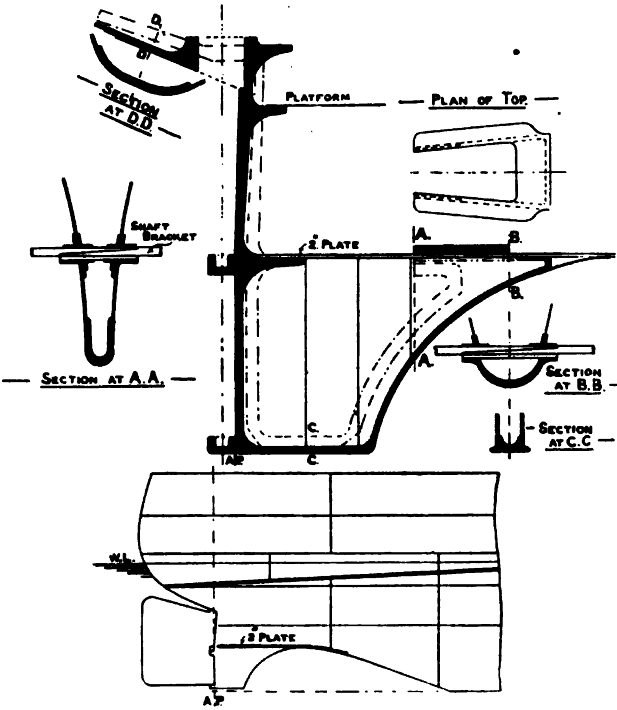


FIG. 70.

the rudder could be supported. In more recent ships the flat portion of the stern, called the "deadwood," has been cut away, as in Fig. 70, and this makes the shape of the sternpost casting rather more complicated. The object of this "cut away" was to maintain good turning qualities; this will be referred to again in Chapter XXI.

Fig. 70 shows in some detail the sternpost of a recent battle-ship, the lower figure giving on a smaller scale the shape of the

Fig. 71.

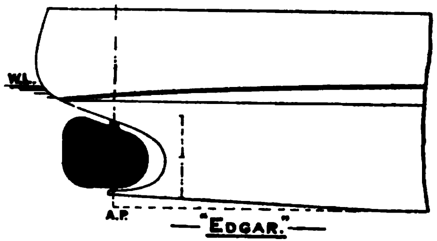


Fig. 72.

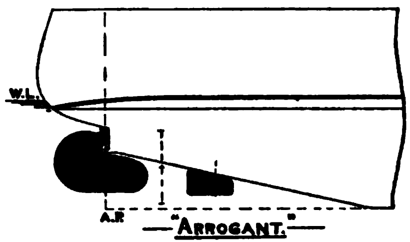
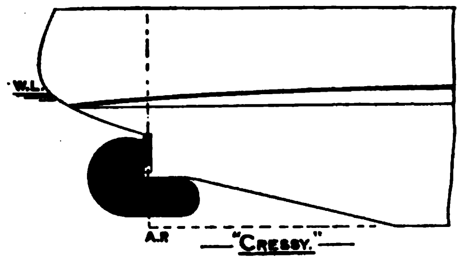


Fig. 73.



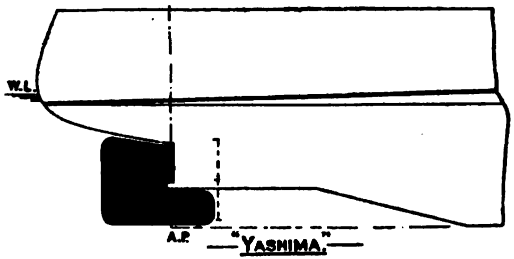


Fig. 74.

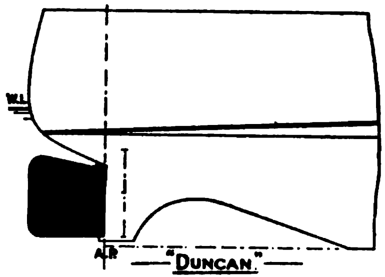


Fig. 75.

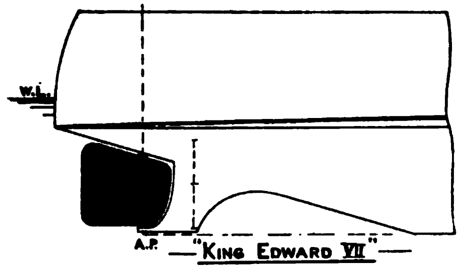


Fig. 76.

Shapes of stern and rudder in various ships.

stern. The keel is cut up about 50 ft. from the stern, but is brought down again to allow about 7 ft. to take the blocks when docking. A 2-in. plate is worked horizontally from the sternpost, extending well forward and securely fastened to the ship's structure. This thick plate forms a substantial bed, to which the lower palms of the shaft brackets and the forward end of the sternpost casting can be secured. It also forms an excellent stiffening to the ship, to take the side bending due to putting the rudder over. The sternpost has projections to take the weight of the rudder, and is swelled out at the upper part to receive the rudder-head. This has to be made watertight by means of a stuffing gland, as in Fig. 69. The casting ends a short distance from the rudder-head, and the shape of the stern is maintained to the upper deck by means of a thick steel plate.

Coming now to the sterns of cruisers, we notice that the stern has been shaped in two ways, in both of which a "balanced" rudder is obtained—

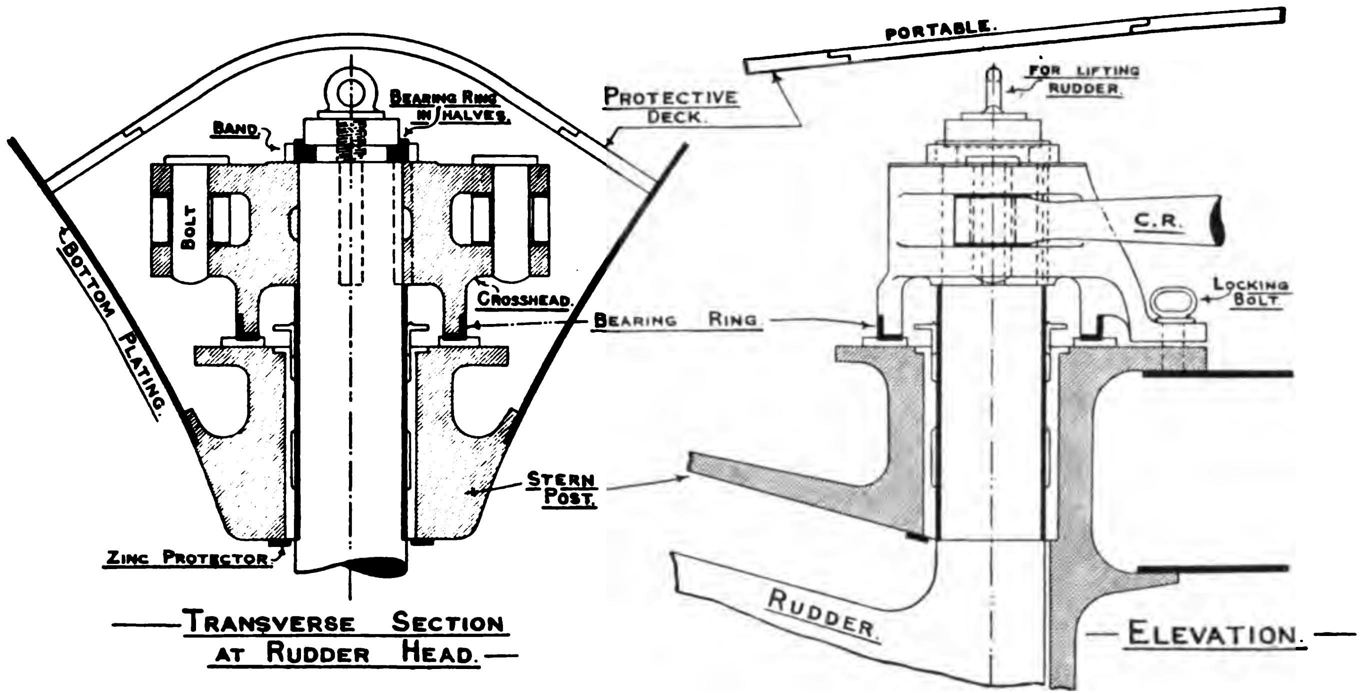
1. As Fig. 71, as adopted in large cruisers up to and including the *Diadem*, and in second and third class cruisers up to the present time.

2. As Fig. 73, as adopted in large cruisers since the *Diadem*. In this type of stern the deadwood is cut right away to facilitate turning, and the rudder is underhung with a portion of the area before the axis.

In either case the weight of the rudder is taken at the top of the sternpost casting, as in Fig. 77, which has therefore to be made specially strong on this account. A steadying pintle is provided at the lower part. Fig. 79 shows in some detail the construction of the sternpost of a large cruiser with underhung rudder.

Rudders.—The shape of rudders in battle-ships up to quite recently has been, as in Fig. 75, of nearly rectangular shape, hinged at the fore side. Fig. 78 shows in some detail the construction of such a rudder. The weight is taken on the sternpost projections, and in order to make the friction as small as possible the bearings are, as shown, of hard steel. The frame of the rudder is a steel casting,[1] lightened out as much as possible, but necessarily of massive construction at the forward end to stand the large twisting moment. The sides are covered with 15-lb. ($\frac{3}{8}$ in.) steel plating,[1] and the space inside is filled with fir. The rudders of the battle-ships

[1] Phosphor bronze in a sheathed ship.



Band
Bearing Ring in halves.
Bolt
Bottom Plating.
Crosshead.
Bearing Ring.
Stern Post.
Zinc Protector.
Transverse Section at Rudder Head.
Protective Deck.
Portable.
For lifting rudder.
C.R.
Locking Bolt.
Rudder.
Elevation.

Fig. 77.

G

of *King Edward VII.* class have a portion of the area before the axis (Fig. 76). This renders the steering of the ship easier, because the centre of pressure on the rudder is brought nearer the axis.

For cruisers rudders are now always "balanced," *i.e.* a portion of the area is before the axis. If we deal with a rectangular plate towed through the water at an angle of 30° to 40°, it is found that the centre of pressure is about one-third the breadth from the

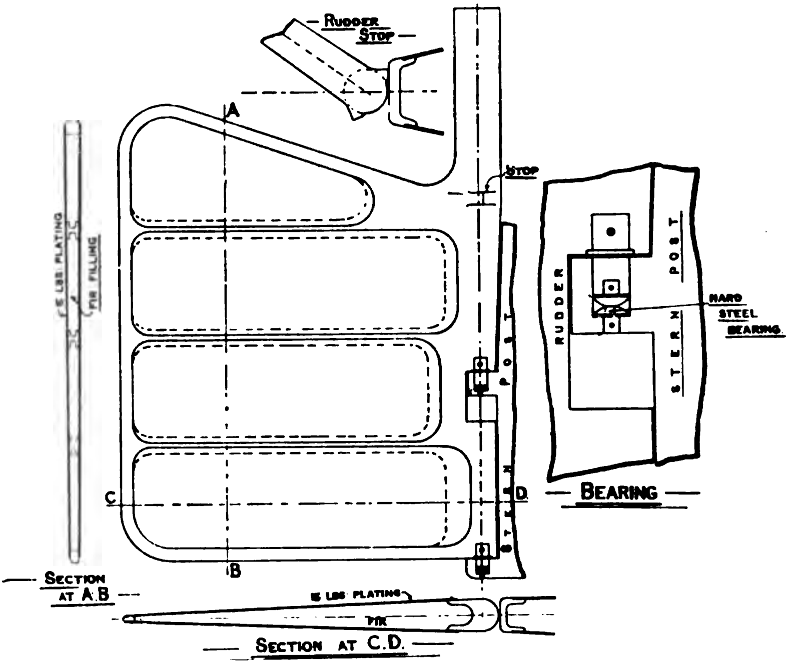


Fig. 78.—Rudder of battle-ship.

leading edge. So to balance a rudder, *i.e.* to get the centre of pressure close to the axis, we need to make the area before the axis considerably less than one-half the total (see Figs. 71 and 73). In such a rudder the twisting moment even at high speeds is small, and much smaller power is needed in the steering arrangements than with an unbalanced rudder. This is specially desirable in cruisers, because of their high speed and the limited room available aft to house the gear owing to the fineness of these ships (see Fig. 163). The pressure, per square foot of rudder area, increases

as the *square of the speed*, so that, comparing 24 knots with $19\frac{1}{2}$ knots, the proportion of pressure for equal areas is $(\frac{24}{19 \cdot 5})^2 = 1 \cdot 5$,

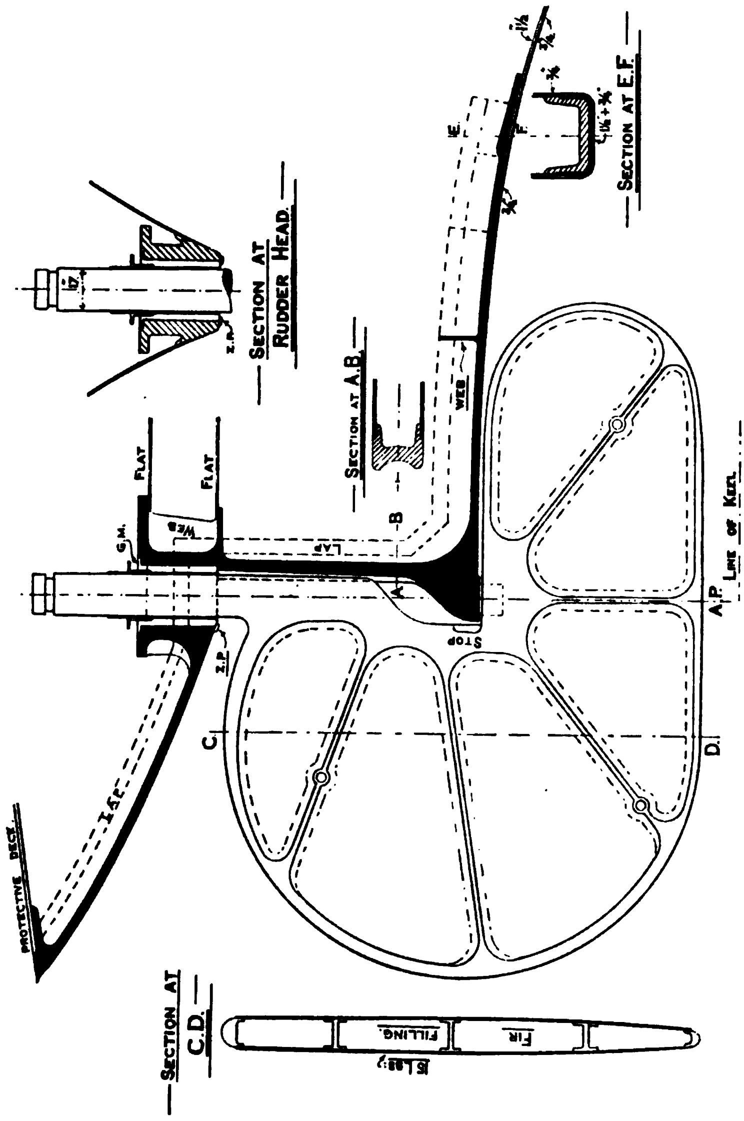


FIG. 79.—Sternpost and rudder of a cruiser.

or an increase of 50 per cent. This represents the increase of pressure to be dealt with in the fastest of our cruisers as compared

with our fastest battle-ships. If, therefore, we fitted a rudder hinged on the forward edge in a cruiser, the steering gear would need to be very massive, the steering engine would have to be of large power, and steering by hand would be difficult. On these accounts the rudders of cruisers are always balanced, so that the moment of the water pressure about the rudder-head is small even at high speeds.

The weight of the rudder is taken by the top of the sternpost, as shown in Fig. 77. At the top of the rudder-head a recess is formed, into which a bearing is placed in two halves. This bearing rests on the rudder cross-head, which has three or four legs. These are connected to a circular bearing ring, which slides in the metal path on the top of the sternpost.

Fig. 79 shows in some detail the construction of the sternpost and rudder of a large cruiser.

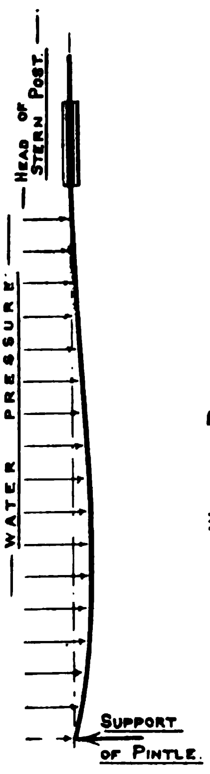


Fig. 80.

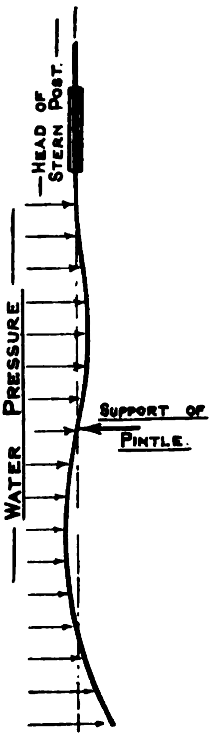


Fig. 81.

In determining the diameter of the rudder-head of an unbalanced rudder, we practically have only the *twisting* to deal with, but in cruisers the *bending* is of large amount, while the twisting going ahead is only small. Figs. 80 and 81 show how the two forms of balanced rudder would bend supposing each is held rigidly in the sternpost. In the first case it is like a beam held at one end and simply supported at the other. In the second case it is held at one end and supported near the middle, with the other end free. In either case a considerable force has to be taken by the lower pintle, and a large bending moment at the rudder-head. The condition going astern has to be investigated, as it may happen that this is the worst case. The maximum speed astern is assumed to be about three-fourths the full speed ahead. The centre of pressure then is nearer the after edge, and the twisting moment about the axis is of considerable amount. This twisting moment, combined with the

bending moment, will determine the necessary size for the rudder-head, unless the ahead conditions require a larger diameter.

In any rudder, the head being under water, it is necessary that the hole in the sternpost should be made watertight. The hole is lined with gunmetal, and the rudder-head is cased with a gunmetal sleeve, as shown in Figs. 69 and 77. A stuffing gland is fitted at the top to make the hole watertight.

Projections are cast on the rudder on each side to bring up against the sternpost when the rudder is hard over; these, however, are being omitted in some recent ships.

It was formerly the practice to supply each ship with a mould giving the actual shape of the rudder. It is the present practice to supply a sketch of the rudder on a large scale, giving complete figured dimensions.

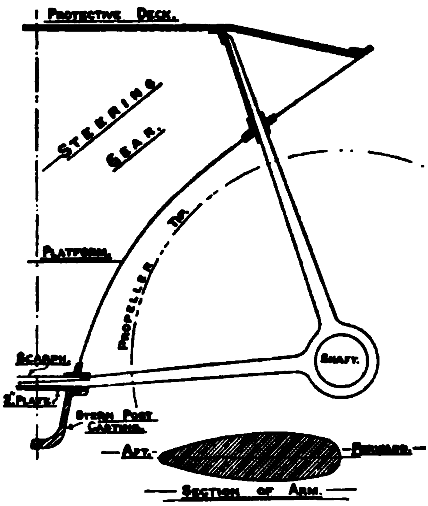


FIG. 82.

Shaft Brackets.—In twin screw vessels a considerable length of the propeller shafting is outside the ship, and brackets are fitted on either side, just forward of the propellers, to take the weight of the after end of the shafting, etc. For steel ships the brackets are of cast steel, for sheathed ships of phosphor bronze. These brackets do not have to take any fore-and-aft thrust (this being taken in the engine-room by the thrust block), but they have to bear the very considerable weight of the propeller, etc. Because of this the attachment to the structure of the ship has to be very secure.

Fig. 82 shows the arrangement in a recent battle-ship. The arms are flattened out at the top and bottom. The upper palm passes inside the ship, where it is riveted to a thick fore-and-aft plate. The lower palm is shaped with a scarph to fit a corresponding scarph on the other bracket. The two brackets are then

securely fastened to the 2-in. fore-and-aft plate, and to the forward end of the sternpost casting (see also Fig. 70). The arms of the brackets are of pear-shaped section, with the blunt end foremost. This is in order to diminish the resistance of these brackets, it having been proved by experiment that this form offers the least resistance when fully submerged. (It will be remembered that a torpedo has a blunt nose and a fine run.)

CHAPTER VIII.

STEERING GEARS.

THE steering gear, fitted at the stern adjacent to the rudder, is a most important fitting of a war-ship, and, except in the smallest classes, is always arranged under water and under protection. The stern has to be specially shaped to do this, and Figs. 71 to 76 show how this is done in different classes of ships. Under ordinary circumstances the steering gear is actuated by the steam steering engine, but hand wheels are provided as an auxiliary.

The twisting moment on the rudder-head (within the ordinary limits, viz. up to about 35°), which has to be overcome by the steering gear, depends on—

(i.) The area of the rudder;
(ii.) The *square* of the speed of the water meeting the rudder;
(iii.) The angle made by the rudder with the middle line, and
(iv.) The distance of the centre of pressure from the axis.

This last is of great importance, and has led to the introduction of balanced rudders, as Figs. 71 to 74, because in these rudders the centre of pressure is close to the axis. This makes the twisting moment small, even at high speeds.

The larger the angle (within the ordinary limits) the greater is the moment to be dealt with. A gear which, with a constant motive force, will operate on a rudder to overcome a large twisting moment at large angles, and a small moment at small angles, will be the most desirable. This is necessarily accompanied by a slower motion at large angles than at small angles. Such a steering gear is termed *compensating.* We shall consider three such gears, which have been largely fitted in vessels of the Royal Navy.

1. **Rapson's Slide Steering Gear.**—This gear is shown in outline in Fig. 83, and in some detail in Fig. 84. It is a very heavy gear, and takes up a lot of room. It has been principally

used in battle-ships. A cross-head on the rudder-head is connected by parallel rods to a second cross-head, on to which a long tiller is attached. The forward end of this tiller is of parallel section, and passes through a block, which can swivel inside another block. This second block is made to travel across the ship on a thwartship path, by means of a sprocket chain. This chain passes to the sides of the ship, and down to the centre line, where it passes under a sprocket wheel, which fits into the chain. This sprocket wheel is made to revolve either by the hand or steam gear, and so the rudder moves as required. This gear takes up a lot of room, because of the travel of the tiller from side to side, and for this reason its use has been confined to battle-ships, which are full at the stern. It has the advantage of compensation, but has the disadvantage of being reversible. If the chain or sprocket wheel broke, the tiller would swing from side to side with the movement of the rudder. On this account a friction brake is fitted, to hold the tiller if necessary. This brake is tightened up to hold the tiller when the gear has to be changed from hand to steam, or *vice versâ*.

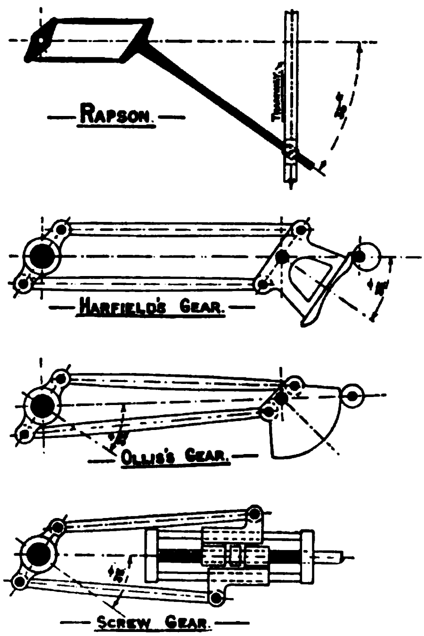


FIG. 83.

Theory of Rapson's slide steering gear.—In Fig. 85 a tiller is shown, passing through a ball, which is made to move along the thwartship path. This is similar to the state of things that obtains in the actual gear. The constraint of the slide brings into action a side force on the ball, as shown, Q. The pull of the chain P, combined with the force Q, gives, by the parallelogram of forces, a resultant force $\frac{P}{\cos\theta}$ acting square to the tiller. This acts at a leverage of $\frac{h}{\cos\theta}$, so that the moment at the angle θ is $\frac{Ph}{\cos^2\theta}$, as compared

with the moment Ph at the middle line. At 35°, $\frac{1}{\cos^2\theta} = 1{\cdot}5$, so that at the extreme angle the moment is 50 per cent. greater than at 0° for the same

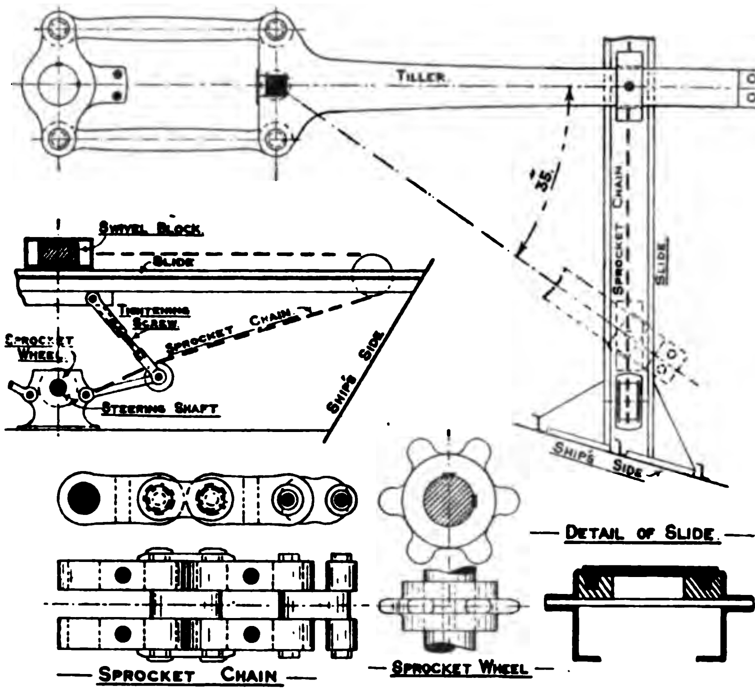


FIG. 84.—Rapson's slide steering gear.

pull on the chain. This increase in moment is accompanied by a slower motion; thus, if a certain number of revolutions moves the tiller through 10° at the start, the same number will move it through about 7° at the end of the travel. The detail of the slide shown in Fig. 84 shows how the side thrust is provided for.

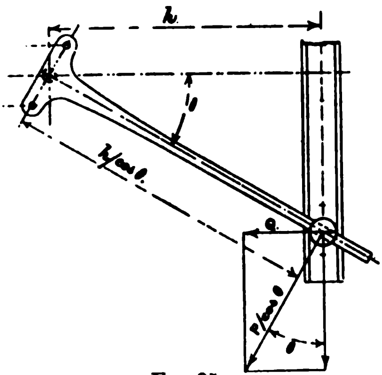


FIG. 85.

2. Harfield's Steering Gear.—This patent gear has been fitted in a number of ships, both battle-ships and cruisers. It has the advantage of compensation, which is obtained as shown in Figs. 83 and 86. The forward cross-head is fitted with a curved rack,

which engages with an *eccentric pinion.* This pinion is made to revolve as required. Assuming a constant moment acting on

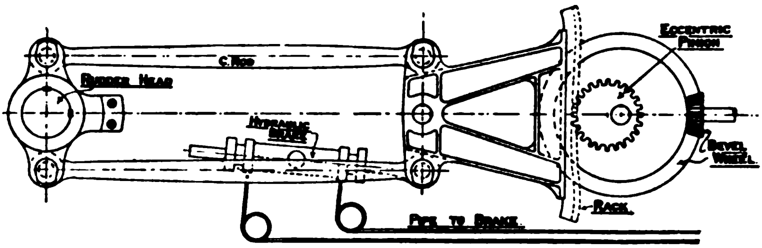


FIG. 86.—Harfield's steering gear.

this pinion, the force on the teeth is greatest at large angles, because the leverage from the turning centre is then smallest.

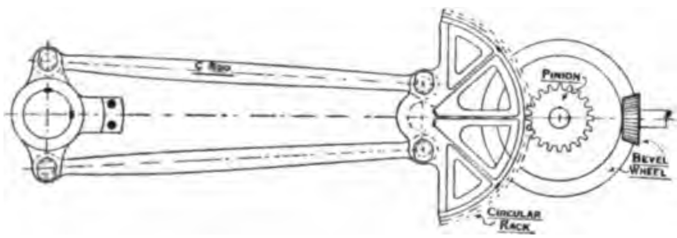


FIG. 87.—Ollis's steering gear.

This large force acts on the cross-head, and at the large angles acts at a larger leverage than at small angles. Thus,

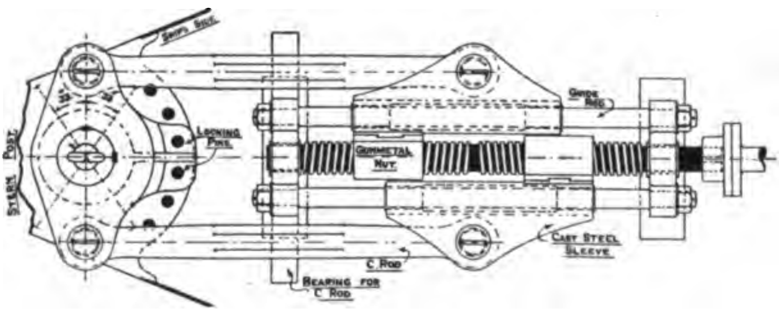


FIG. 88.—Screw steering gear.

at the extreme angles we have a greater moment on the rudder obtained in this double way, a larger force on the teeth, and a

larger leverage. Thus the compensation desired is obtained. With this gear a hydraulic brake is fitted, which can be made to hold the rudder in any desired position. The safety of the gear depends absolutely on the teeth of the rack and pinion in contact, and on this account these are made of forged steel of ample strength. The gear is reversible, like the Rapson's slide, unless a worm and worm-wheel is introduced, which has been done in some ships.

3. **Ollis's Steering Gear** (Figs. 83 and 87).—This is a gear designed at the Admiralty by Mr. F. B. Ollis.[1] It answers the same purpose as Harfield's gear, with simpler construction. The pinion and rack are both circular, but the connecting rods to the rudder cross-head are brought close together at the forward end. Assuming a constant moment on the pinion, it is seen that as the angle increases the connecting rods come closely together, and so the force along these rods increases, giving a larger moment at large angles than at small. This is the compensation desired. The gear is reversible, as in the previous cases, and it is necessary to provide a friction brake, to hold the gear in case it becomes disabled, and when changing from hand to steam, or *vice versâ*.

4. **Screw Steering Gear** (Figs. 83 and 88).—This gear is quite different to the previous gears, being non-reversible, and being the reverse of compensating. It has, however, been very largely adopted. It consists of a right- and left-handed screw, which, on rotating, causes two nuts to slide along in opposite directions; these nuts are prevented from rotating by being made to slide along parallel bars. Connecting rods are led from these sliding nuts to the rudder cross-head.

The gear takes up very little room. Another great advantage in this gear is that it *cannot reverse*, and the rudder in any position is locked. It is because of this that it is being now fitted to all the new ships of large size in the Royal Navy. It is, however, the reverse of compensating, and a rudder not balanced would be difficult to move by the hand wheels at large angles when a high speed is reached. All the large ships at present building are, however, being fitted with rudders more or less balanced, and in these ships the screw steering gear should prove a satisfactory fitting. A friction brake has usually been fitted on the rudder cross-head, to hold the rudder in case anything has to be done to the gear.

[1] Now Chief Constructor at Hong Kong.

Steam Steering Gear.—Whatever steering gear is adopted at the stern, the motive power for working it under ordinary circumstances is a steam steering engine. Formerly the engine was placed aft, just forward of the steering gear, but this is objectionable for the following reasons:—

(i.) The steam-engine heats the compartment, making special ventilation necessary.

(ii.) The steam and exhaust pipes led from the boilers heat the compartments at the after end of the ship, through which they pass.

(iii.) Attention of the engine-room staff is required at the engine.

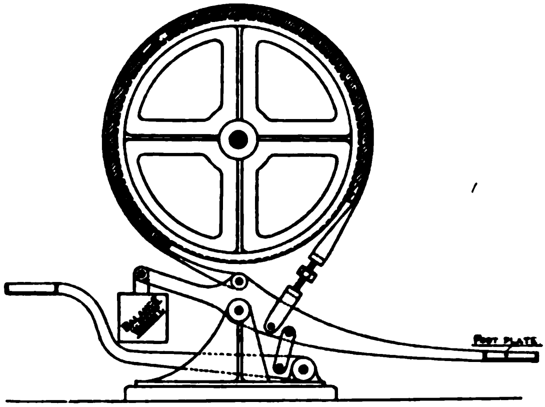
In recent ships the steering engines, of which there are two, are placed on the after bulkhead of the engine-room, and shafts are led along the shaft passages to the steering gear. This entails a good deal of weight, but on the whole is more satisfactory than the former system.

The steam steering engines are of ample power, and in a large ship the time specified to take the rudder from hard-over to hard-over is only 30 seconds when the vessel is going full speed. A worm and worm-wheel is fitted between the shaft and the steering engine, so that shocks from the rudder are kept from the engine.

The steam engine is operated by means of steering wheels placed as follows in a large ship:—forward bridge, forward conning tower, after conning tower, lower deck forward in the ammunition lobby, platform aft in the steering compartment. The engine is worked from these positions by means of controlling shafting. This shafting is led down inside the armoured tubes from the conning towers to below the protective deck. Care is taken to avoid any rigid connection of the shafting to the protective deck or any portion of the vessel exposed to damage in action. Some ships have been fitted with telemotor controlling gear instead of the shafting. In this gear the motion is conveyed to the steering engine from the steering wheel by means of small pipes. A small piston is worked at the wheel; this causes motion of water throughout the system, and exactly similar motion at the steering engine.

Steering by Manual Power.—Most vessels have one position below, aft, for steering by manual power, in case the steam gear is not available. The hand wheels are four in number in the largest vessels, three and two being fitted in smaller ships. In

any gear not self holding, any shock on the rudder is transmitted to the hand wheels, and considerable power is required to hold



FAYRER'S BRAKE.

FIG. 89.

the gear at any required angle. To hold the hand wheels in such a gear a "Fayrer's brake" is fitted. This is a band brake bearing

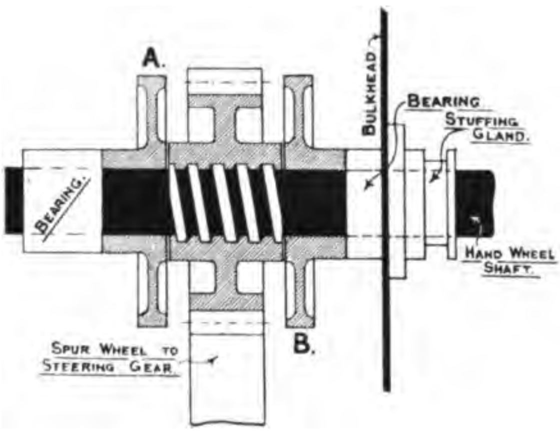


FIG. 90.

on a wheel on the hand-steering shaft. It can be tightened by standing on a foot-plate either side (Fig. 89).

In Harfield's gear an interesting clutch is fitted just abaft the hand wheels (Fig. 90). The wheels A and B are loose on the

shaft, and are provided with pawls at A and B, so that motion of A in a right-handed direction and motion of B in a left-handed direction (looking from the bulkhead) is impossible. The pinion, which gears with the spur wheel on the steering gear, moves endwise on the shaft by means of a thread, the shaft acting as a bolt and the pinion as the nut. If the hand wheels are rotated in either direction it will be noticed that the pinion is drawn along to the wheel A or B, which is free to rotate. If, however, the motion is reversed, *i.e.* the rudder becomes the motor in either direction, the pinion is drawn along to, and jams against, the wheel A or B, which is prevented from rotating, so that the rudder cannot react on the hand wheels.

Such a clutch or a Fayrer's brake is unnecessary when a worm and worm wheel is introduced into any portion of the gear, as this renders the gear non-reversible.

Spare Gear.—With such an important fitting as the steering gear, a large margin of strength is provided in all the parts, and spare parts are provided for certain portions of the gear, so that these may be replaced in case of disablement. The following is the list of spare gear provided in a recent vessel fitted with screw gear. A similar list is provided for other gears.

One connecting rod, two bushes for same, two pins for same; one spur wheel 78¼ in. diameter; one spur wheel 45½ in. diameter, one bush for same; two nuts for screw shaft keep plates; two sets of brasses for screw shaft; one gunmetal nut, right-handed; one gunmetal nut, left-handed; one set of brasses for guide rods, No. 4. All of these are tried in place and stowed.

In some gears a length of shafting in the shaft passage is made the weakest part of the gear, so that this would be the first to go, and so save the steering gear itself. A spare length of this shafting is carried to replace.

Clutches, etc.—All steering wheels are so arranged that the upper part of the wheel moves in the same direction as that of the ship's head. In a large ship the steam steering wheels take the rudder from hard-over to hard-over in eight complete turns, the hand wheels requiring twenty-four complete turns.

Clutches are fitted to enable the gear to be changed quickly from connection with one engine to the other, or from either engine to the hand wheels. In doing this it is necessary to hold the gear (unless this is self holding), by means of a brake, usually a friction brake. Indicators are provided in a prominent position,

showing the angle at which each engine, hand wheels, and the steering gear itself are situated. **It is of the greatest importance that connection should only be made when the engine or hand wheels, as the case may be, indicates the same angle as the steering gear itself.** For, suppose the engine at 5° port is coupled with the steering gear at 0°, then when the steam-engine is taken over to hard-a-starboard the gear will bring up against the stops with the engine at full speed, and some part of the gear will be fractured. In recent ships the steam gear brings up at 35°, the gear itself at 37°, and the rudder at 38°. The clutches to the steering gear should be so arranged that it is not possible for the steam steering engines to become geared up together, or for either to be geared up to the hand wheels under any circumstances.

Auxiliary Steering Gear.—It was until recently the practice to provide auxiliary means of steering, for use supposing the steam-engines and hand wheels were disabled. This usually consisted of a set of blocks and tackles, which could be fastened to the gear to work it by hand. The gear provided entailed considerable weight, and was difficult to rig up in place. Ships which had the auxiliary provided never used it, and it has been removed from a number of ships. The present practice is simply to rely on the hand wheels as the alternative in case the steam-engines are not available. From this point of view, the use of twin screws is an alternative method of steering, and cases have occurred recently in which ships have had to go long and stormy voyages without a rudder at all, the steering all having been performed by the twin screws.

CHAPTER IX.

PUMPING, FLOODING, AND DRAINAGE.

THE present chapter deals with the methods adopted in ships of the Royal Navy for clearing water out of the ship, which has entered through damage or has accumulated in the bilges under the ordinary conditions of working; and also the means for voluntarily admitting water into certain portions of the ship to preserve stability, or to keep the ship upright or on an even keel after damage, or to flood the magazines, etc., in case of fire.

By *Drainage* is meant the means of allowing water to pass from one compartment to another until it reaches a pump-suction by which it can be removed.

By *Flooding* is meant the deliberate admission of water into the ship through a sea-cock. This may be necessary to correct heel or trim after damage, or in the event of fire.

By *Pumping* is meant the general arrangements adopted to remove water from the ship by means of the *fire and bilge* pumps in the engine-rooms, or by the *Downton* pumps, worked by manual power.

Before dealing generally with the subject, it will be advisable to see how far the pumps fitted to a ship are able to deal with the inflow of water.

If A be the area of a hole in a ship's bottom in square feet, and d the distance of the centre of the hole below water, then the initial velocity of the water through the hole is given by $8\sqrt{d}$ ft. per second, so that every second there would be $8A\sqrt{d}$ cubic ft. of water entering, or about $14A\sqrt{d}$ tons per minute. If the hole is 16 ft. below the surface, about 56 tons will enter in a minute for every square foot of area.

The following is the specified capacity in tons per hour of the pumps available for dealing with such a leak in a recent large ship, viz.—

Four *centrifugal pumps* (used ordinarily for circulating water through the condenser drawing from the sea), which can draw from the engine-room bilge	5600 tons
Four *fire and bilge pumps*...	400 tons
Four *Downton pumps*	100 tons
	6100 tons

That is about 102 tons per minute. But we have seen above that the water through 1 square ft. amounts to 56 tons a minute, so that a hole only 2 square ft. in area (19 in. diameter), 16 ft. below surface, is sufficient to overpower the whole pumping capacity of a large ship. Such a hole is small in comparison with what would result from a collision, and this illustrates the importance of efficient watertight subdivision, which has been dealt with in previous chapters. By this means the damaged compartments may be isolated, and a collision mat placed in position over the hole so that repairs of a temporary nature may be undertaken.

In recent ships the arrangements for pumping and drainage are considerably simpler than formerly adopted; it is proposed only to deal with the arrangements in a battle-ship of the *Duncan* class, as being typical of recent practice.

Main Drain.—Special arrangements are made for clearing the ship of large quantities of water for which the ordinary steam and hand pumps would be inadequate. For this purpose a *main drain* (Fig. 91) is worked above the inner bottom from the forward boiler-room, branching to either engine-room. This drain is 15 in. diameter in the middle boiler-room, and 20 in. × 15 in. in the after boiler-room. By means of this drain, water can be passed from either boiler-room into one or both engine-rooms. Sluice valves, with non-returns, are fitted as shown to control the flow of water to the engine-rooms; the non-return valves automatically prevent the passage of water from the engine-rooms or from one boiler-room to another. In each engine-room, suctions are taken down from the *circulating pumps* of the condensers, so that these pumps can be made to draw from the engine-room bilge if desired instead of from the sea. These circulating pumps work independently from the main engines, and their engines are placed as high in the engine-rooms as possible, in order that they may continue working even if the engine-room is flooded to a considerable depth. The sluice valves on the main drain are worked from below and also from the main deck.

In addition to the main drain, the forward compartments likely to have large quantities of water are drained into the forward

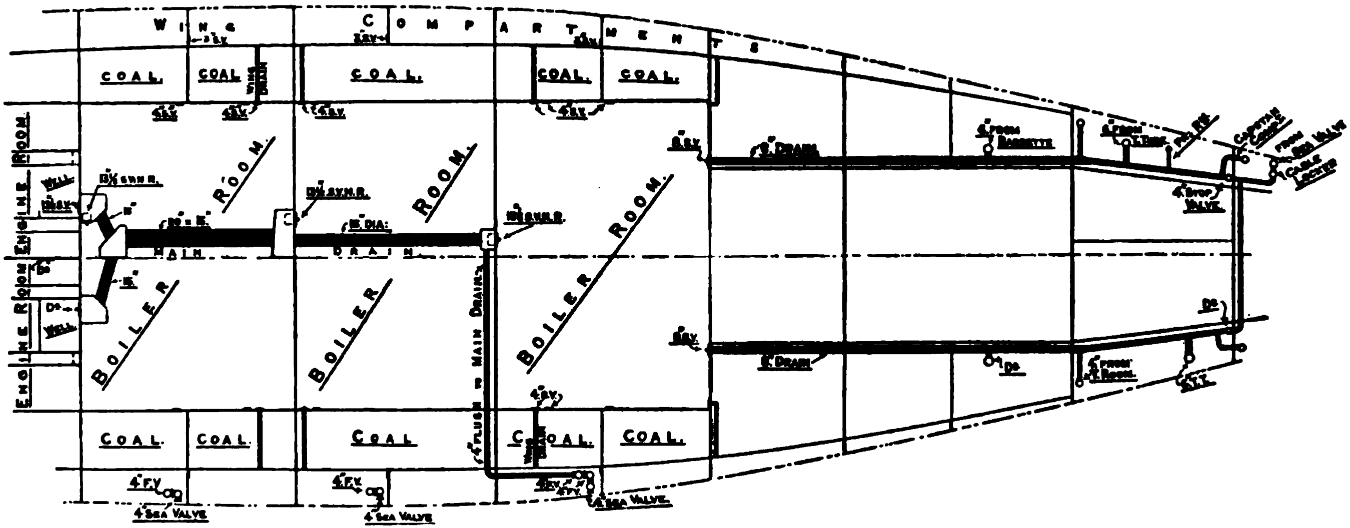


FIG. 91.—Main drain, etc.

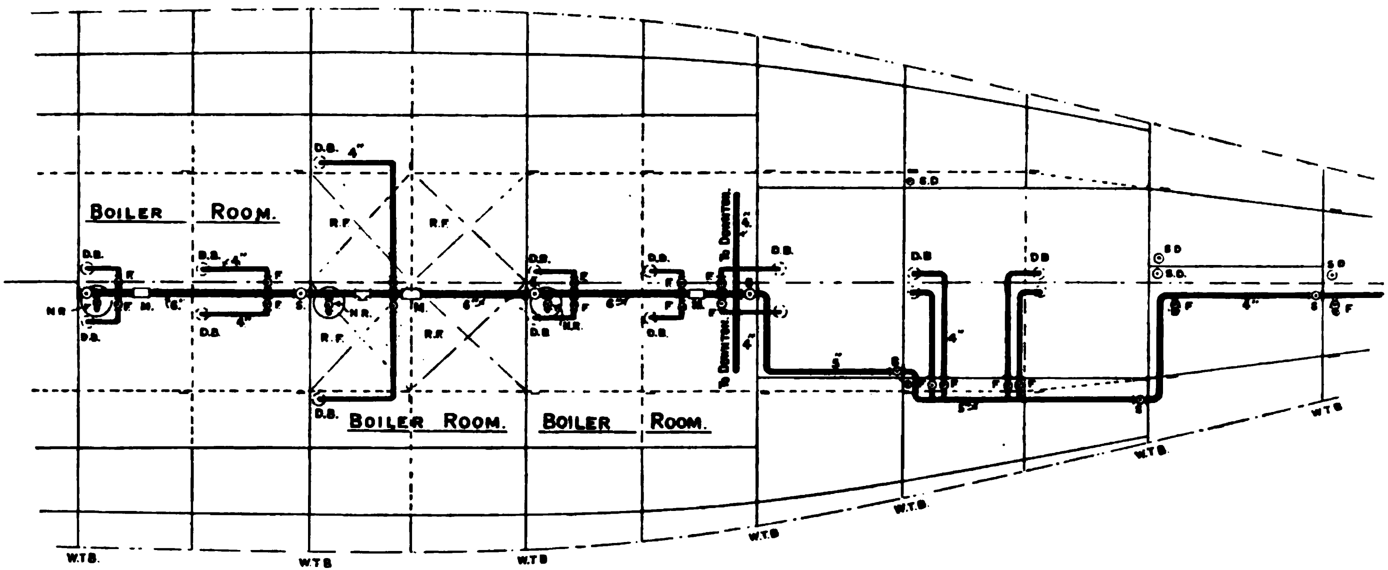


FIG. 92.—Main section.

boiler-room (Fig. 91). Similar drains are fitted aft, draining into the engine-rooms. At the forward end we notice that two pipes, 8 in. diameter, are used, and into these we have pipes leading from barbette, submerged torpedo-room, torpedo-tubes, chain lockers, etc. The forward and after drains, and also the main drain, each have connection to a sea valve for flushing purposes. The wing

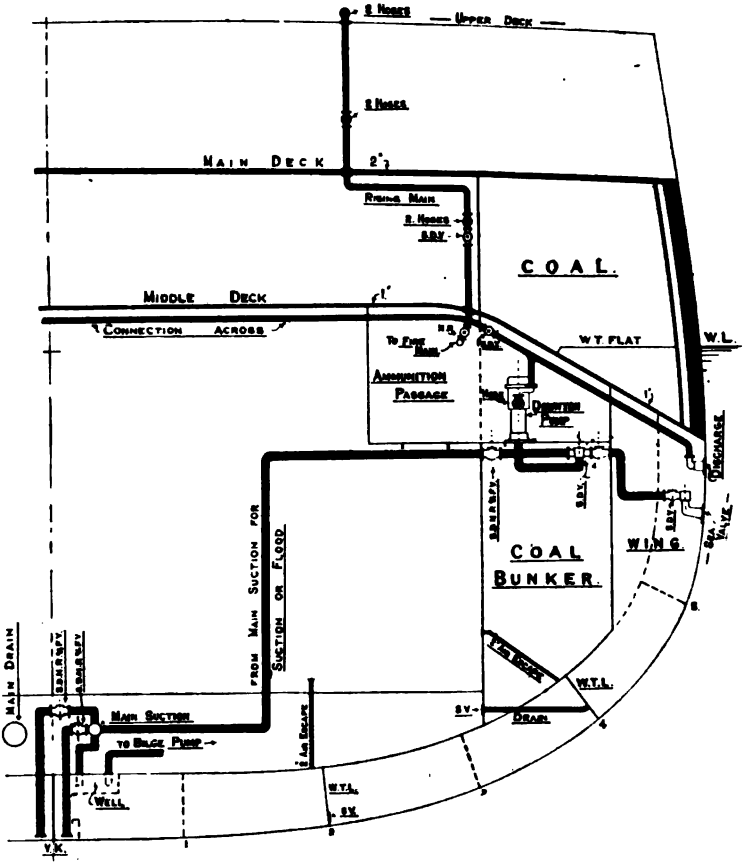


FIG. 93.—Pumping section.

compartments and inner coal-bunkers are drained on to the top of the inner bottom, as shown in Figs. 91 and 93.

When a flat requires draining, a valve, as Fig. 99, is fitted. Above the flat there is a strainer with screw-down valve, and below the flat a flap valve to act as non-return.

Pumping.—For freeing the ship of ordinary quantities of water, a *main suction* is fitted above the inner bottom, extending nearly the whole of the ship's length. Screw-down valves, marked S in Fig. 92, are fitted on this main suction at each main transverse bulkhead in order to preserve the watertightness of the bulkhead in the event of the pipe being damaged on either side. Fig. 92 shows a portion of this main suction. Pipes are taken from it to the double-bottom compartments, and on each of these branches a valve, F, is fitted, which under ordinary circumstances acts as a *screw-down non-return* valve. The valve, however, can be unlocked and lifted off its seat, when it will allow the compartment to be flooded (see Fig. 97 for construction of valve). The double bottom between Nos. 2 and 4 longitudinals is drained through S.V. on No. 2 longitudinal, as shown in Fig. 93, to the suction at the middle line (except in way of the portion of the double bottom used for the reserve feed water). Suctions are also led from the main suction to shallow pockets in the engine and boiler-room bilges; these suctions have a S.D.N.R. valve, without the flooding arrangement shown in Fig. 97. A suction is also taken from these pockets, independent of the main suction, direct to the steam pumps. Mud pockets, M, are placed at intervals on the main suction as shown. This main suction also has branches to the bottom of the ship, forward and aft of the double bottom, as shown in Fig. 92. The main suction is connected with the steam pumps in the engine-room, so that any desired compartment may be pumped out by the use of the steam pumps of the ship.

Downton Pumps.—A connection is also made with each of the four 9-in. Downton hand pumps. These are placed in recesses in the ammunition passages (Fig. 93). Each Downton is geared to work from either the main or middle decks, as most convenient, by means of long cranks, on to which a number of men can be placed.

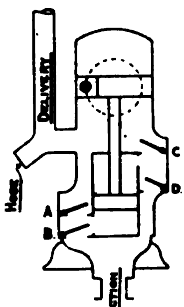


FIG. 94.—Downton pump.

The *Downton* is a double-acting pump, shown in outline in Fig. 94. There are four valves, A, B, C, and D, all opening upwards. A piston works up and down as shown. Suppose water is in the tail pipe and the piston comes down. This causes B to close and A to open, so that air is forced up the left-hand side of the pump. This action creates a partial vacuum above the piston, and water

will rise and fill the upper part of the piston chamber. During the upward movement this water is forced up through the valve C, the valve D closing. The water which then collects below the piston is forced away during the downward stroke. In this way a continuous flow of water is delivered either overboard through the discharge or into the rising main (Fig. 93).

Fig. 95 shows in diagram the various leads from a Downton pump. There are three valves, viz.—

1. A screw-down (S.D.) valve connecting to the Kingston valve (K) (see Fig. 98 for construction of a Kingston).

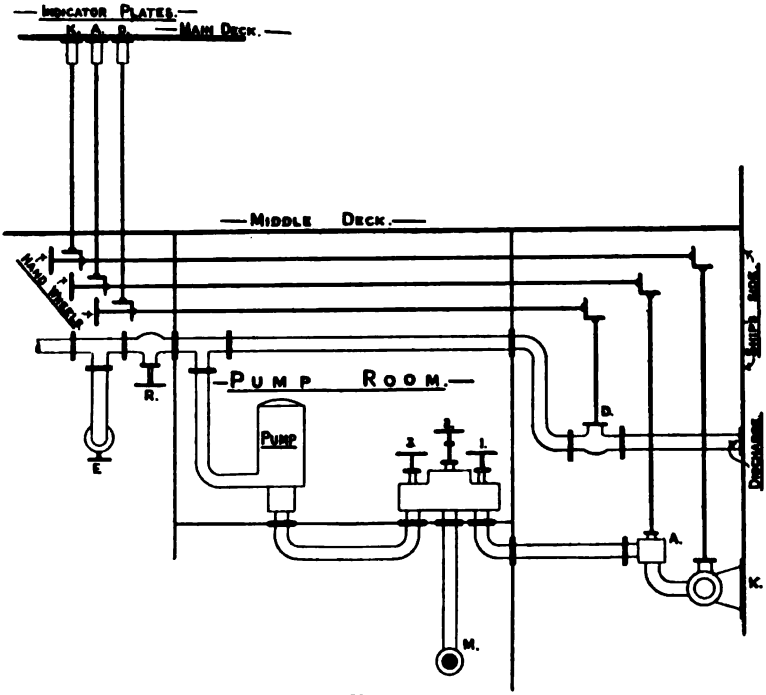


FIG. 95.

2. A screw-down non-return and flood (S.D.N.R. and F.) valve (Fig. 97) connecting to the main suction (M).

3. A S.D. valve connected to the tail pipe of the pump.

The pump has—

(i.) A discharge overboard through S.D. valve (D).

(ii.) A discharge into a rising main with a branch into the fire main.

The branch to the fire main has a N.R. valve, E, so arranged

that when the fire-engines are working the pressure from the fire main is prevented from affecting the hand pumps or their system of pipes. The rising main from the Downton is shown in Fig. 93, and this can be used for wash deck purposes, etc., hose connections being fitted as shown. A connection is made across the ship to the pump on the opposite side. Fig. 95 shows how the valves at the ship's side are geared to be worked from the level of the pump, and also from the main deck if desired.

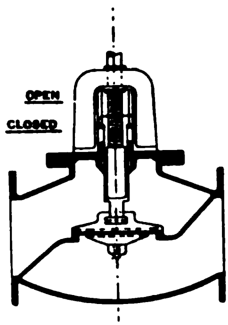


FIG. 96.—Screw-down valve.

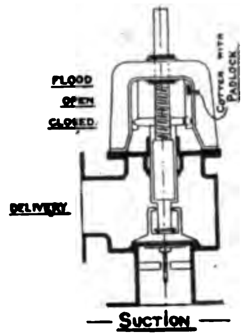


FIG. 97.—Screw-down non-return and flood-valve.

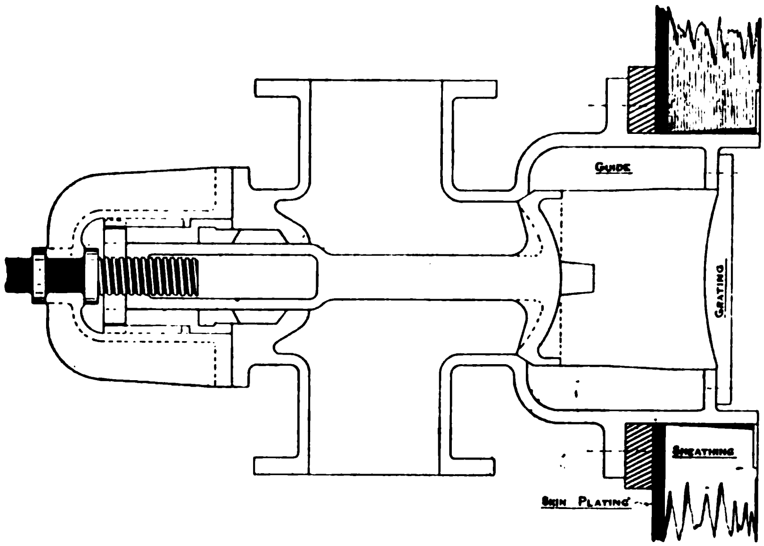


FIG. 98.—Kingston valve to a sheathed ship.

To pump any compartment out, all the valves between the fire-engine or Downton and the compartment must be opened. Thus for a forward compartment all the bulkhead stop valves

must be opened between the pump and the compartment, besides the valve on the particular branch.

A book is issued to each ship, and, among other things, it is stated how each compartment may be drained or pumped, and where the valves are worked from. A specimen page is given at the end of this chapter. Take, for instance, the *forward boiler-room*. It can be drained into the engine-room through the main drain; the 13½-in. S.V. is worked near the valve, and also at the main deck, the deck-plate being between stations 71 and 73 on the port side. The boiler-room can also be pumped out through the branch from the main suction. The 4-in. S.D.N.R.V. on this branch is opened from a deck-plate on the middle deck between stations 71 and 73.

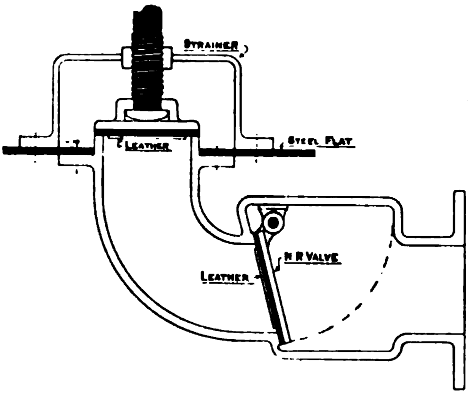


FIG. 99.—Non-return valve.

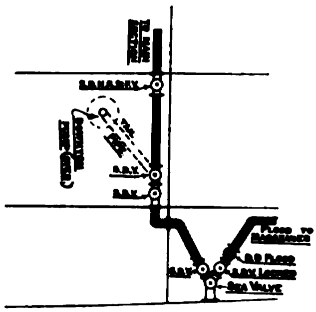


FIG. 100.

In "protected" cruisers suctions are taken down to the bottoms of the upper bunkers to clear water away. These ships would get water above the protective deck if the thin side were riddled.

In working a valve it should be noted that the valve is not necessarily closed when the indicator points "shut," as after some wear of the gearing it is necessary to work beyond this point to get the valve well down on its seat.

Flooding.—In order to permit flooding of the compartments of the double bottoms, the valve 2 in Fig. 95, and the valve to the particular compartment, both of which under ordinary circumstances of pumping act as *non-return* valves, can be lifted up off the seats so that flooding can take place. This, however, can only be done deliberately, as the valve cannot be lifted right up unless a locked pin is withdrawn (see Fig. 97). In order,

therefore, to flood a double-bottom compartment we have to open a number of valves (Fig. 95), viz. Kingston (K) valves A and 1, unlock and lift up valve 2, and also unlock and lift up the valve leading to the particular compartment. The book above-mentioned states from which Kingston each compartment is flooded.

The wing compartments are flooded direct from sea valves placed as shown in Fig. 91. It will be noticed that each pair of wings has a flooding valve. The trimming tanks forward and aft can also be flooded direct from the sea.

Special arrangements are made for flooding magazines direct from the sea. Usually one Kingston supplies a number of magazines, etc. Besides the valve at the Kingston a valve is placed at the magazine, both of these being *locked valves,* which must be deliberately released to allow flooding to take place. They can be worked from the main deck or at the valves as may be desired. For trial purposes the outer valve would be first opened so that the connecting pipe would be filled. This valve is then closed and the magazine valve opened. The water which then enters can be caught in a bucket. Where a Kingston valve is used only for magazine flooding, it is only necessary to fit one locked valve close to the magazine in addition to the Kingston. Where a Kingston serves other purposes a valve must be fitted at the Kingston, so that it may be open without flooding the magazine flooding-pipe (see Fig. 100). No means are provided for draining magazines.

Dry Dock Flood.—Fittings are also provided for flooding magazines when the ship is in dry dock. A pipe is taken, between the Kingston and the magazine (Fig. 100), to the upper deck, and fittings, to take hose connections, are placed on this pipe above the main and upper decks. In the event of fire, hoses could be taken from the dockyard and the magazines flooded. In recent ships the hoses from dockyard are arranged to connect on to the Kingston direct, by means of a fitting which can be secured to the Kingston opening when the ship is in dry dock.

Air Escapes.—When flooding any closed compartment it is necessary to provide an escape for the air. Pipes are led from the crown of the compartment, as in Fig. 93, with plugs on the upper ends. These plugs, when unscrewed slightly, allow the air to escape, and when the compartment is full they can be readily screwed up again. Plugs are also provided on the manhole covers (Fig. 48).

The air escapes from the magazines are constructed with a

simple lift valve, as Fig. 101; the pipe is led well up above the crown of the magazine and bent round, with the end perforated. In ships in which the crown of the magazine is near the waterline, these air escape valves are provided with a spring to cause them to lift easily under the small pressure.

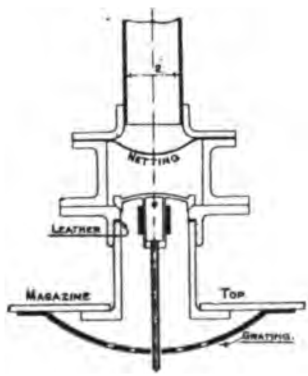


Fig. 101.—Air escape to magazine.

Fire Main (Figs. 102 and 103).—The fire main and its system of pipes is obviously a most important set of fittings in a war-ship. The fire main itself in a large ship is a copper pipe 5 in. diameter, running all fore and aft under the protective deck. It is connected in the engine-room with the fire and bilge pumps, stop valves being fitted to shut off the fore-and-after portions of the system as desired. The fire main system can be charged from the Downton pumps if required, but the connection is fitted with a non-return valve, so that the pressure in the fire main may not affect the Downton pumps. The fire main is not rigidly connected to the protective deck, but is either supported from bulkheads or from slings connected to the deck. Branches are taken from the fire main to each stokehold, to submerged torpedo-rooms, capstan engine-room, ammunition passages, etc., with connections at each place for attaching a couple of hoses.

Rising mains are led at intervals to the upper deck, two of these are led up under protection inside the barbettes. Each rising main has a stop valve beneath the protective deck at its junction with the fire main, and these valves are placed in accessible positions so that they can be worked from below the deck if desired. They are also geared to work from above the deck. To most of the rising mains there is also a stop valve beneath the main deck, to shut off the pipe above that deck in the event of the pipe being damaged in action. Branches are taken from the rising mains for flushing w.c.'s, washing out ash-shoots, washing out barbette guns, etc. One-inch bib valves are placed on rising mains between middle and main decks, main and upper decks, and in ammunition passages for drawing off small quantities of water if desired.

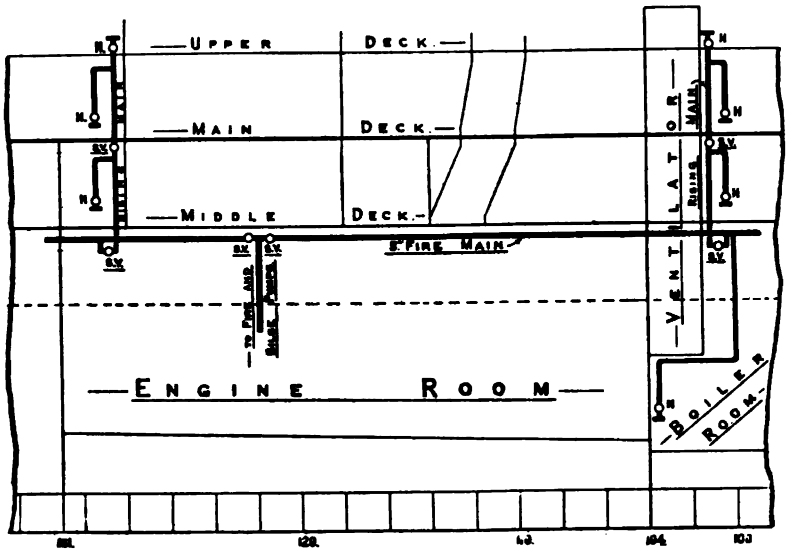


Fig. 102.

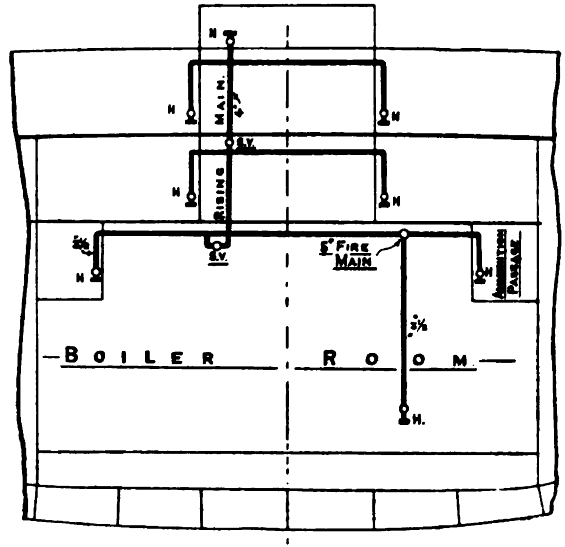


Fig. 103.

Bulkheads of compartment.	Name.	Capacity in cubic feet.	Decks bounding compartment.	Means of access.	Position for working watertight doors.	How drained and pumped.	Position for working drain valves.	How flooded.	Position for working flooding valves.	How ventilated and valves how worked.
39—47 starboard	Wing compartment	680	Middle deck, fourth longitudinal	Manholes 39—41, 45—47	Engineers' stores, L.D.; hydraulic engine-room, L.D.	Drained into compartment aft by 3-in S.V.	Lower deck 47—49	—	—	—
39—47 port	Ditto	680	Ditto	Ditto	Engineers' workshop, L.D.	Ditto	Ditto	—	—	—
47—57 starboard	6-in. shell-room	2,750	Platform, inner bottom	Hatch 55—57	Light room on platform	—	—	—	—	*Supply*, 5-in. branch with S.V. from "G" fan chamber 55—57 *Exhaust*, through hatch
47—57 port	12-pounder shell-room	2,020	Ditto	Ditto	Ditto	—	—	—	—	*Supply*, 4-in. branch with S.V. from "G" fan chamber 55—57 *Exhaust*, through hatch
47—57 starboard	Small arms magazine	1,800	Ditto	Trunk with hatches 55—57	Ammunition lobby, L.D.	—	—	Through "A" Kingston and flood valve; then through 4-in. flood valve 49—51	A. Pass. and main deck 55—57 for Kingston, etc. Hyd. room and main deck 49—51 for flood valve	*Supply*, 5-in. branch with S.V. from "G" fan chamber 55—57 *Exhaust*, 5-in. S.V. with A.V. handwheel to supply and exhaust in A. Pass.
47—57 port	3-pounder magazine	1,800	Ditto	Ditto	Ditto	—	—	Through "B" Kingston and flood valve; then through 4-in. flood valve 49—51	A. Pass. and main deck 55—57 for Kingston, etc. Engineers' workshop and main deck 49—51 for flood valve	Ditto
47—57 starboard	Wing compartment	1,000	Middle deck, fourth longitudinal	Manholes 47—51, 55—57	Ditto	Drained into fore boiler-room through 4-in. S.V.	Forward boiler-room on 57 bulkhead	—	—	—
47—57 port	Ditto	1,000	Ditto	Ditto	Ditto	Ditto	Ditto	—	—	—
57—73	Forward boiler-room	34,000	Middle deck, inner bottom	W.T.D. on 73 bulkhead starboard; also by ventilation trunks	At door and on main deck 71—73 starboard. Doors in trunks, main deck, 61—63 starboard; 67—69 port	Drained into main drain through 13½-in S. and N.R.V. Pumped through 4-in. S.D.N.R.V. on main suction	Near valve and at main deck 71—73 port. Middle deck 71—73 for suction	—	—	*Supply*, special trunks and fans *Exhaust*, uptakes, etc.
57—65 starboard	Lower coal-bunker	1,940	Ditto	W.T.D. 57—59; also escape to lower deck	At door and on main deck 57—59; escape at ammunition passage 63—65	Drained into forward boiler-room through 4-in. S.V. 63—65	Near valve	—	—	*Supply*, 6-in. pipe with louvre at upper deck 63—65 *Exhaust*, 6-in. pipe with louvre at upper deck 67—69

CHAPTER X.

VENTILATION.

The problem of the effective ventilation of a ship is a more difficult one than the ventilation of a building. In the case of a building, ventilation is assisted because the porous nature of the walls allows air to diffuse through, this *diffusion* being all the more effective as the difference of temperature within and without is greater. In the case of a ship, however, the conditions are quite different, and the impervious nature of the skin renders diffusion through impossible; ventilation must be obtained by the actual introduction of fresh air and the withdrawal of the foul air.

Ventilation can be either *natural* or *artificial.* By natural ventilation is meant supply and exhaust without the aid of fans. By artificial ventilation is meant that fans are used to draw pure air down to, or draw foul air away from, the space to be ventilated, or both. Natural ventilation is used for the ventilation of the 'tween deck spaces, and for the special cases of the coal-bunkers and spirit-room. Artificial ventilation is used for the other spaces below, as magazines, shell-rooms, store-rooms, etc.

For the 'tween decks cowls are used which can be turned to face the wind, the exhaust taking place through the hatchways, and in some cases through other cowls which can be turned away from the wind. The side-lights are also useful for ventilation purposes. For the spirit-room it is necessary to have an independent supply and exhaust, with cowls on the upper deck, as it is undesirable to have any connection between the spirit-room and any other system of ventilation which might communicate with a magazine.

Coal-bunker Ventilation.—The efficient ventilation of coal-bunkers is of extreme importance, because of the gas that comes from the coal. This gas when mixed with air forms an explosive mixture, and if it is allowed to accumulate may cause serious

explosions. The gases are especially liberated from the coal, if a sudden fall of the barometer occurs, or if the temperature rises.

It is worth noting in this connection that it is undesirable to take wet coal on board, because the moisture causes a rapid and dangerous generation of heat and gas. The coal should also be kept as dry as possible (thus when cleaning the main deck the coaling scuttles should be kept closed.) Temperature tubes are provided in all coal-bunkers, so that the temperature in the body of the coal may be ascertained at frequent intervals. No light except a safety lamp must be used inside a coal-bunker until it is ascertained that the bunker does not contain explosive gas. Special precautions in this respect are necessary for a few days after coaling.

In the ventilation of a coal-bunker two pipes are employed, one for the introduction of fresh air and one for the withdrawal of the foul air. The latter is led where possible up a funnel casing. These casings under ordinary circumstances will be hot, and this causes the air in the pipe to rise. This induces a current of air over the top of the bunker, and fresh air is then drawn down the supply pipe leading from the upper deck.

There are slight differences in the fittings of different ships, but the general principles will be understood by Fig. 104, which shows the ventilation of the coal-bunkers of a battle-ship. In this type of ship there are three separate series of bunkers, which are each independently ventilated.

1. The *upper bunkers*, behind the armour, which extend over the length of the double bottom.

2. The *lower bunkers*, abreast the engine- and boiler-rooms.

3. The *wing bunkers*, abreast the engine- and boiler-rooms (can be used for coal if desired, and in the *deep load* condition these are assumed to be filled).

1. *Upper bunkers.*—These bunkers are formed into groups, Fig. 102 shows the arrangements for the forward group of four. A trunk is taken into the funnel casing into which all the four bunkers exhaust. Each bunker has a separate supply by means of a pipe leading from the upper deck. The coaling scuttles in the main deck can also be removed and the gratings put in if desired.

2. *Lower bunkers.*—The supply pipes for these bunkers are taken down the ventilators in order to avoid piercing the thick decks, and the exhausts are taken up the funnel casings. A separate supply and exhaust is taken from each bunker, the supply pipe being taken to the far corner so as to get the current of air well over the top of the bunker.

3. *Wing bunkers.*—These bunkers are ventilated similarly to the lower bunkers.

The upper end of each supply pipe is fitted with a louvre with fittings for locking, and the louvre has engraved upon it, "*Not to be closed without special order.*" The top ends of the exhausts are led above the upper deck, each with a mushroom top and a throttle

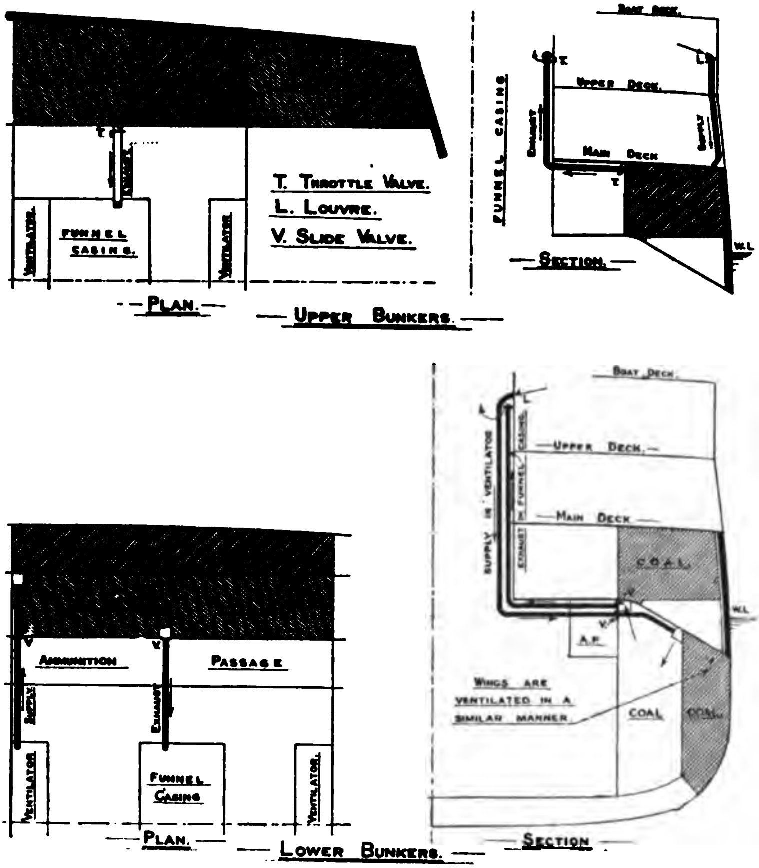


FIG. 104.—Ventilation of coal-bunkers.

valve for closing. Where these pipes have a bend where water is likely to accumulate, a plug is fitted so that the water can be drained away. The flanges of the deck beams have holes as necessary to allow air to circulate freely over the top of the coal when the bunker is full. The slide valves and throttle valves to

the supply and exhaust pipes have been omitted in recent ships. Also the exhausts end in a louvre on the side of the casing.

Double Bottoms, etc.—It is important to note that care must be taken in dealing with confined spaces which are closed for long periods. In such spaces poisonous gases are likely to accumulate, and nobody should be allowed to enter until it has been ascertained that the air is pure enough to breathe. This can be done by putting a lighted candle in for a few minutes. The manholes to such spaces are in pairs (see Fig. 49), fitted as far apart as possible, and hand fans can be used to ventilate the spaces before entering.

Ordinary Ship Ventilation.—1. *With large steam-driven fans.*—For the artificial ventilation of spaces below, the system adopted, up to and including *Canopus* class, was by means of a number of large steam-driven fans. Trunks were led from these fans, and these trunks were pierced by louvres to allow air to pass into the compartments through which the trunks pass. Fig. 105 shows some specimen leads in a ship of the *Majestic* class. In these ships there were ten fans, eight being 6 ft. in diameter, two being 4 ft. 6 in. in diameter. Some of the most important of these are connected together, so that if a fan breaks down the other can do its work. With this system the watertight bulkheads are constantly pierced by the non-watertight trunks, and it is necessary at such places to fit valves which will automatically close the opening. These valves are designed so that if water rises on either side of the bulkhead or floods the flat, a float is lifted which releases the valve closing the opening. Two forms of these *automatic valves* have been largely adopted, viz. Beck's and Broadfoot's (Figs. 63 and 64). In Beck's valve an ordinary slide valve is used, which is connected to a balance weight. When the float rises it releases this balance weight, and the valve then shuts. In Broadfoot's valve, the valve turns round and closes the opening. In either case a small hole is made in the bulkhead with a pipe leading to the float, so that if water rises on the opposite side to the float, the valve will be shut. This hole is closed by the movement of the lever carrying the balance weight. Small pipes are led from wells in the main or upper deck, so that in case of emergency the valves can be closed by pouring water into these wells.

There are objections to this system, viz.—

(i.) The large fans require steam-engines, which heat the compartments, and which require long lengths of steam and exhaust pipes.

(ii.) The use of the automatic valves to keep the flats and

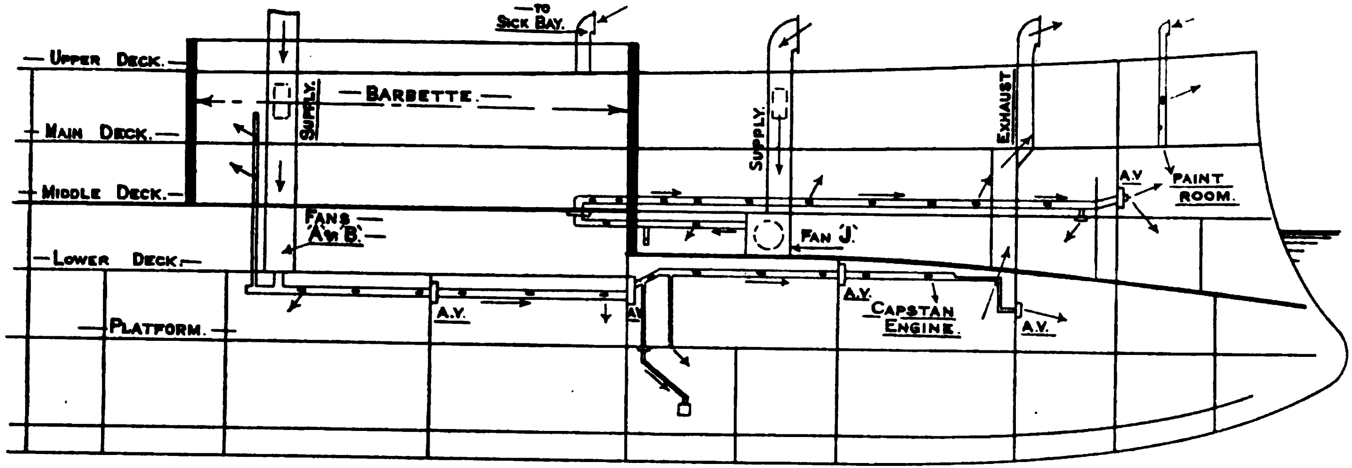


Fig. 105.—Ventilation with steam fans.

bulkheads intact. These are frequently found either to work too stiffly, so that they do not act, or to be too sensitive, so that the motion or vibration of the ship continually closes them. In the former case the watertightness of the bulkheads is completely destroyed, and in the latter great inconvenience is caused by the shutting off of the ventilation.

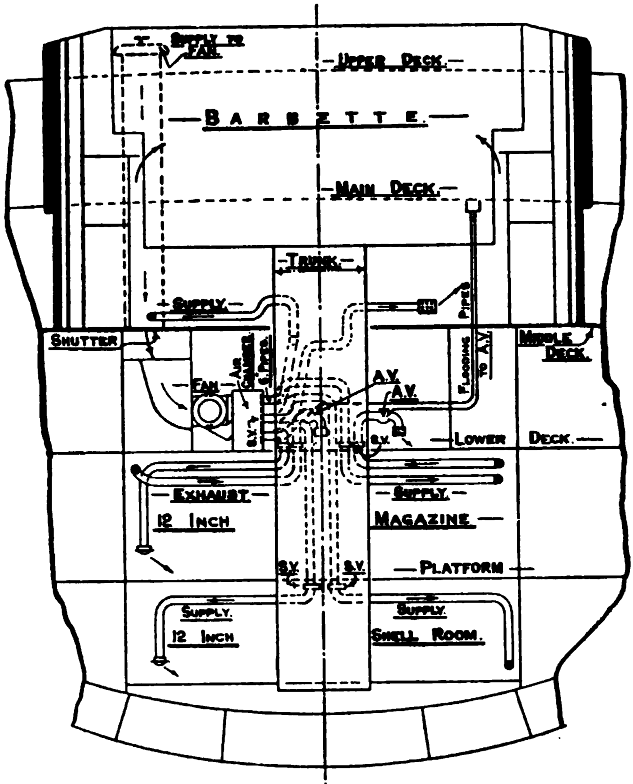


FIG. 106.—Ventilation with electric motor vans.

In the above system of ventilation the exhaust is usually obtained through hatchways and doors, but for some compartments this is insufficient, and special exhaust trunks are led to the upper deck with cowls. Such spaces are compartments for auxiliary machinery, capstan engine, and steam steering engine. (These trunks can be used for escape purposes.)

The special arrangements for magazine ventilation will be referred to later.

2. *With small fans driven by electric motors.*—The system now adopted is to have a large number of smaller fans driven by electric motors taking the current from the electric light circuit of the ship. The principle of the system is that the vessel is divided into a number of sections, so that the main watertight bulkheads of the ship below water are not pierced,[1] and the compartments in each of these sections are supplied with air by one or more motor fans. In this system automatic valves are unnecessary, and the inconvenience of heating by steam-engines and steam-pipes is obviated.

In a recent large ship there are fifteen of these fans, 24 in. in diameter. Fig. 106 shows a set of leads to and from a specimen fan. The fan draws air from a trunk opening above the upper deck, with a mushroom top. It delivers the air into an air chamber, and from this six pipes are led. All these pipes are closed by a single slide valve, worked at the valve or from the main deck. These pipes are watertight and lead—two to the barbette, two to the 12-in. shell-room, and two to the 12-in. magazine. The pipes into the magazine and shell-room have slide valves at the entrance. The top of the supply trunk can be closed at the upper deck, and a sliding shutter is also provided at the middle deck.

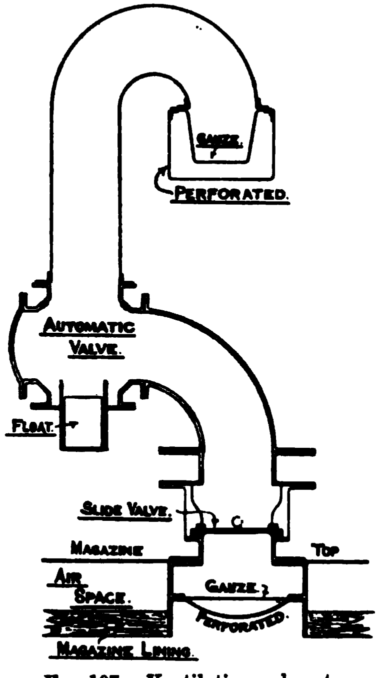


Fig. 107.—Ventilation exhaust to magazine.

For ordinary compartments, as in the previous system the exhaust is obtained through hatchways and doors, but for some, special exhaust trunks are necessary, as seen above.

In either of the above systems of ventilation, spaces like

[1] Some of the pipes ventilating spaces above water are led through watertight bulkheads. These have watertight slide valves at the bulkheads, but not automatic valves.

provision-rooms, etc., are not supplied with separate ventilating pipes, but they can be ventilated when required by means of a hose taken to a nozzle on an adjacent ventilation pipe.

Magazines.—Magazines require special arrangements for ventilation, because of the presence of the ammunition. It is necessary to avoid any undesirable rise of temperature, and also any excessive amount of moisture, because the ammunition is very sensitive in these respects. It must be possible that ventilation can proceed when the magazine is closed. For this purpose, in addition to the supply, special exhausts are fitted to the crown of the magazine; two of these are shown to the 12-in. magazine in Fig. 106. The detail of an exhaust is shown in Fig. 107. It has a slide valve, and in addition an automatic valve, which closes if the flat is flooded. These automatic valves can also be closed by flooding pipes leading from the main deck (Fig. 106).

In some ships the magazines have been placed at the middle line of the ship, between the boiler-rooms. Here, in addition to ordinary ventilation, a continuous current of air has to be maintained in the space between the magazine and the boiler-room, in order to keep the magazine cool.

The book of watertight compartments supplied to each ship (of which a specimen page is given at the end of Chapter IX.) gives particulars of how each compartment of the ship can be ventilated.

CHAPTER XI.

CORROSION AND FOULING.

Rust.—If bare iron or steel is allowed to remain in contact with moist air containing carbon dioxide (CO_2), a chemical action goes on, by which the oxygen in the gas unites with the iron, and forms certain oxides of iron. We term the resulting compound *rust.* A similar action goes on if the iron or steel is immersed in fresh or salt water, owing to the carbon dioxide contained in the water. Rusting action is much hastened by heat, and where heat and moisture exist together, rusting goes on very rapidly indeed. Rust is about six times as bulky as the iron from which it is formed.

Corrosion.—Corrosion is very much accelerated by galvanic action. If iron and copper are immersed in dilute acid and metallic connection is made between them, an electric current is set up, and the energy of the current is provided at the expense of the iron. If iron and zinc are similarly immersed, it is found that the zinc wastes away. Not only do different metals act in this way, but different parts of the same plate may be sufficiently apart in the electro-motive series, owing to differences of density, etc., to give rise to an electric current if immersed in dilute acid. Iron and its rust are sufficiently different as to give rise to a current, and it is the iron which becomes corroded, so that when rust is once formed it does not cover up and protect the material, but itself is a cause of further corrosion. Rust is *hydroscopic, i.e.* it takes up moisture, and so dampness gets between it and the iron, and hastens the rusting and corrosion.

A black oxide is formed on steel during the process of manufacture, called *mill scale.* This stands in the same relation to the steel as ordinary rust, in that it is electro-negative to the steel, giving rise to an electric current, by which the steel is further corroded. This mill scale clings very tenaciously to the steel, and

does not come off by ordinary scraping. It is, however, most necessary that it should be completely removed before any paint is applied, and this is done in Admiralty practice by the process termed *pickling*. The plates before being worked are immersed for a few hours on edge in a bath of dilute hydrochloric acid (1 part acid to 19 parts water). This loosens the scale, and the plates when removed are brushed with hard wire brushes, and washed with a hose to remove all traces of the scale and acid. The portions of the structure thus pickled are those liable to come into contact with sea or bilge water, viz. outer and inner bottom, lower plates of bulkheads, plates of fresh-water tanks, and plates of frames.

Steel is more liable to suffer from corrosion than iron, and as the steel is thinner than corresponding iron of the same strength, a given amount of corrosion is relatively more serious in a steel structure than in iron. Small vessels are usually built relatively much stronger than large vessels, because of the necessity of providing a margin against corrosion. In the case, however, of vessels of special construction like torpedo-boat destroyers, the conditions are such that the weight of the hull structure must be the least possible consistent with sufficient structural strength, and the thickness of the plating, etc., is in consequence not enough to leave much margin against corrosion. Thus in these ships the greatest care and frequent examination are necessary, in order to guard against corrosion in its early stages. In these vessels the plating, etc., is *galvanized* before painting, *i.e.* after being thoroughly cleaned the material is immersed in a bath of molten zinc. The layer of zinc thus deposited is a valuable preventive against corrosion.

It is clear from the above remarks that no part of the structure of a ship must be allowed to remain bare, but all must be securely protected against rust and corrosion. A point of considerable importance is the necessity for the thorough cleaning and drying of the iron or steel before applying any paint, as if any rust or scale remains the corrosion will go on underneath the paint, and if the surface is not dry the paint will not adhere. In building H.M. ships it is laid down that each portion as finished must be thoroughly cleaned and painted with linseed oil or thin red-lead, to prevent oxidation while building. Subsequently three coats of red-lead paint are applied, except in confined spaces, where oxide of iron is used. The object of these successive coats is to

hermetically seal the surface of the steel. In store-rooms, etc., the first coat is of red-lead, and the two succeeding coats white.

Wherever any fitting, made of a copper alloy, like gunmetal,[1] is attached or adjacent to the steel structure, an electric current is set up, as already stated, if the paint comes off the steel. The necessary condition for this is that there shall be metallic connection between the copper and the steel, both being immersed in dilute acid (sea

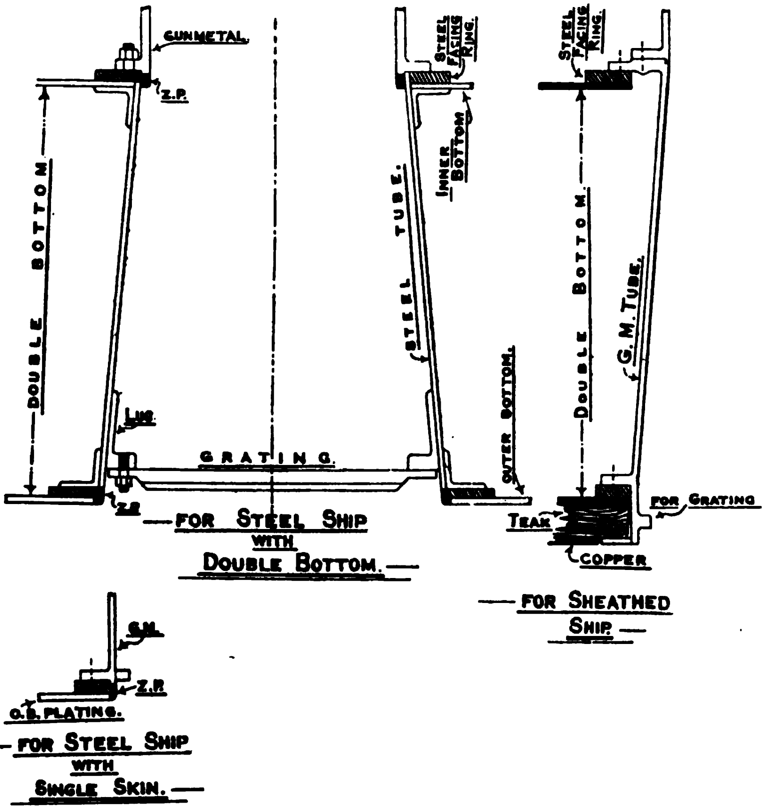


FIG. 108.—Zinc protectors, etc.

water or bilge water is acid enough for this purpose). Under these circumstances very rapid pitting would take place in the plating, etc., in way of gunmetal valves, rudder-head, propellers, etc., and in all such places a strip of zinc is secured to the plating, called a *zinc protector* (see Figs. 77 and 108 for examples). (For a

[1] Gunmetal has the composition—copper 88, zinc 2, tin 10.

sheathed ship it will be noticed that these zinc protectors are not necessary (Fig. 108)). The current then goes on between the gun-metal and the zinc, rather than between the steel and zinc or the steel and copper, and the zinc becomes corroded. The zinc thus protects the steel structure by its own decay, and it can be readily examined and renewed as opportunity offers. It is evident that these *zinc protectors should be left unpainted.*

Fouling.—In dealing with the outer surface of a ship below water, we have not only to seal up the plating to prevent corrosion, but also to avoid, as far as possible, the attachment of marine growths and animals. This is termed *fouling,* and some means of preventing fouling must be adopted in order to keep the resistance as low as possible. The increase of resistance due to fouling is very considerable, and if a certain speed is desired it means a larger expenditure of I.H.P., with consequent increase in coal consumption, or with a given I.H.P. a serious loss of speed.

Copper is the most efficient anti-fouler, and in vessels which have to serve on distant stations with the likelihood of remaining afloat for long periods without docking, it is necessary to use copper sheets to prevent fouling. The action of copper in this regard is held to be as follows: The action of sea water on copper is to form certain copper salts. These form a deposit which dissolves off the surface, and any marine growths, etc., which have attached themselves to the ship are thus washed off as the ship moves along. This action of copper is termed *exfoliation.* Although the copper is poisonous, it is not by poisoning that the growths, etc., become detached, since these only use the ship to attach themselves to, the nourishment being obtained from the water. There is, however, said to be some poisonous effect on germs that are deposited.

With wooden ships, therefore, for many years copper was the standard material to avoid the attacks of marine animals, etc., and to prevent fouling. When, however, iron came into use for ship-building, it was at once found that copper was impossible because of the galvanic action between the copper and the iron skin unless they were insulated from each other. For ships on isolated stations and in tropical waters, the *composite* system of construction was largely employed. In this system the framing was of iron or steel, and the skin was formed of two thicknesses of wood. On to this the copper sheets could be nailed, and insulation

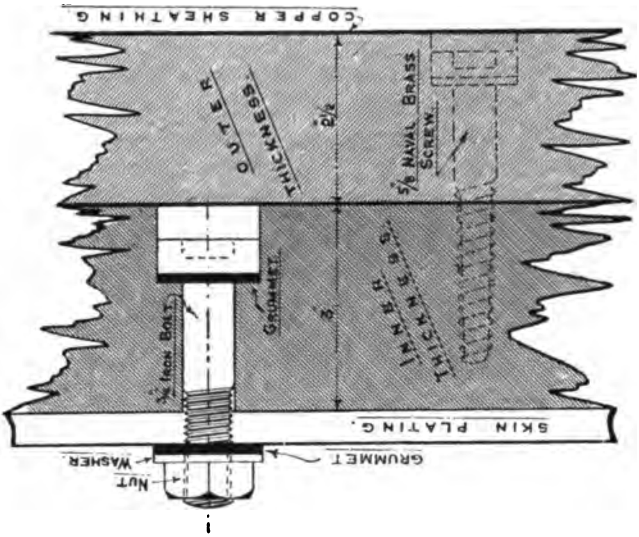


Fig. 110.

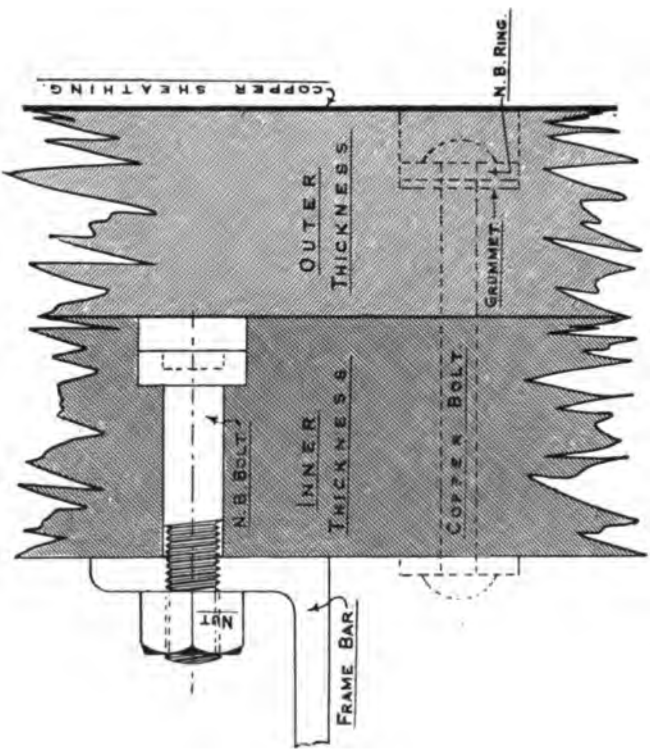


Fig. 109.

between the copper and the structure was obtained. Fig. 109 shows the method of securing the wood sheathing. The inner thickness was connected to the frame bars by naval brass[1] bolts screwed through, with a nut on the inside. The outer thickness was connected to the inner by copper through bolts. The composite system is not, however, structurally strong enough for vessels of any size and power, and no ships of the Royal Navy have been built on this system for some time. The system adopted instead has been to build the ship completely like an iron or steel ship, with skin plating as usual; but on to this skin wood sheathing, usually teak, is fastened. We thus get a ship structurally as strong as an iron or steel ship, with the

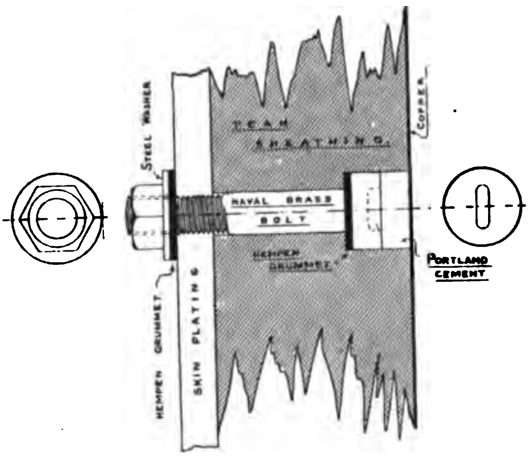


FIG. 111.

advantage of the copper sheathing. It is obviously of the highest importance to have complete insulation between the copper and the iron or steel skin, and in the most common method adopted up to 1887, two thicknesses of planking were used, fastened as shown in Fig. 110. The inner thickness was secured to the plating by galvanized iron bolts, and the outer to the inner by naval brass screws, care being taken to keep the points away from the skin. The system was expensive, and in practice it was found that water found its way between the planking and the plating, and the iron bolts became corroded. The *Calliope* was

[1] Naval brass is composed of copper 62, zinc 37, tin 1.

sheathed on this system, and when she was taken in hand in 1891 for refit and repair, the outer thickness of planking was removed, the inner thickness refastened with naval brass bolts, and the outer thickness refastened.

The system adopted since 1887, for sheathed vessels in the Royal Navy, is by a *single thickness of teak*.[1] The principal features of this system are as follows:—

1. The adoption of such a thickness of wood sheathing as will admit of thorough caulking. The mean finished thickness of teak accepted is 4 in. for large ships, and 3½ in. for the smaller classes.

2. The use of naval brass bolts and nuts with their points *screwed through* the bottom plating, and with plate washers underneath the nuts.

3. The thorough water-testing of the skin plating before the planking is worked.

4. The most careful fitting, fairing, and fastening of the planks; the coating of all faying surfaces with suitable compositions before the planks are fitted in place, and the subsequent injection of composition in order to fill any spaces left between the planking and the plating.

5. The use of hempen grummets steeped in red-lead, under the bolt heads and under the plate washers to secure watertightness in the bolt holes. The use of a plug of cement over the bolt head to prevent the sheathing nails coming into contact with the bolts and destroying the insulation.

Fig. 111 shows the method of fastening above described.

For any vessel sheathed with copper it is impossible to use iron or steel for stem, sternpost, etc., in consequence of the galvanic action referred to above. For these vessels these portions are made of the copper alloy *phosphor bronze*, a material which can be made to give good castings, and which also possesses good strength.

Copper sheathing.—The surface of the vessel is payed over with pitch, and tarred paper is placed on, and the copper sheets are then fixed on with brass nails.

Ships of the largest size have been built on this system, and for vessels on foreign stations likely to remain undocked for long periods, the prevention of fouling obtained is worth the extra expense involved. The following comparisons show the cost in

[1] See paper by Sir W. H. White (I.N.A., 1896).

money and measured mile speed in the case of ships of *Edgar* and *Apollo* classes. In these classes a direct comparison is possible, as the vessels were similar in all respects except in the matter of sheathing.

	Navy List displacement.	I.H.P.	Measured mile speed (designed).	Additional cost due to sheathing.
				£
Unsheathed *Edgar* .	7,350	12,000	20·0	
Sheathed *Edgar* . .	7,700	12,000	19·7	17,000
Unsheathed *Apollo* .	3,400	9,000	20·0	
Sheathed *Apollo* . .	3,600	9,000	19·75	10,000

For the outer bottom plating of a steel ship we have to rely on anti-fouling compositions to keep the bottom clean. With vessels running on fixed routes at uniform rates of speed, experience should soon determine the composition which gives the best results. For vessels of the Navy, however, such uniform conditions do not obtain, and compositions which are suited to a vessel at anchor for long periods may be of little use to the same vessel when steaming. Trials are continually being made to determine the most suitable compositions to be used, and to investigate the merits of new paints offered for trial.

The usual basis of these anti-fouling compositions has been copper, but because of the galvanic action between copper and iron or steel, it is held that copper is undesirable. The introduction of poisonous matters as copper or arsenic is said to have an effect on the germs of marine growths that are deposited. An anti-fouling paint should have a certain soapiness, so that by wasting away slowly it may get rid of the marine growths, etc., that attach themselves, and by exposing the poisonous matters kill the germs that are deposited. It is clear, therefore, than an iron or steel ship requires frequent docking in order to renew the anti-fouling paint. On these occasions the bottom should be carefully examined to see if the surface has become corroded.

Prevention of Corrosion Inside.—We have seen above that no part of a ship's structure must be left bare, or else rusting and corrosion will certainly ensue. Examination is continually necessary to ascertain how far the paint is protecting the steel structure, and it is necessary to provide access to all parts for this purpose. For places of which no use is made, access is usually

obtained by manholes, Figs. 48 and 50, the latter being the type fitted on bulkheads, etc., and the former to the double-bottom compartments. The means of access to every compartment of a ship is given in the book of watertight compartments supplied, of which a specimen page is given at the end of Chapter IX.

It is laid down that every accessible part of the outer and inner bottom and framing is to be inspected once a quarter by the engineer officer and the carpenter, and any defects discovered are to be made good. In the event of dampness, the steel must be thoroughly dried and all traces of rust removed before applying the paint. *Well-slacked* lime is to be used in places from which water cannot be removed.

Every three years (annually in the case of destroyers and ships whose plating does not exceed ¼ in.) a complete survey is made by the dockyard officers.

Pipes at the lower parts of a ship should preferably be of galvanized iron, and not copper or lead. If, however, copper or lead pipes have to be used, it is necessary that they be well painted, covered with canvas, painted to make quite waterproof. In way of metal valves zinc protectors are fitted to assist in preventing corrosion.

The inner bottom plating under engines and boilers is specially liable to corrosion, especially the upper surface, due apparently to the fretting action of the ashes and hot water. These parts should be frequently examined, and where rust is found to be forming, or where the paint is abraded, the surface should be thoroughly scaled, cleaned, and dried, and coated with three coats of red-lead paint.

Cement.—The cementing carried out in recent ships is of far less extent than that formerly adopted. The double bottom, which under ordinary circumstances will not contain water, is not cemented at all except in those spaces used for reserve feed water, where the bottom is coated with hard cement 1½ to 2 in. thick. Before and abaft double bottom, just sufficient cement is used so that water will not obtain a lodgment anywhere, but will readily flow to the pump suctions. In parts, as at the extreme ends of the ship, where a considerable amount of cement is necessary, the cement is mixed with coke to keep the weight as small as possible. Cement may even be detrimental, supposing it to get cracked through any cause, as then water will get down to the plating, and corrosion may go on unnoticed.

The insides of fresh-water tanks are coated with "Rosbonite."

In living spaces, etc., corrosion and discomfort is caused by the *sweating* of steel work, owing to the condensation of moisture from the air. In such spaces the under side of decks, bulkheads, etc., are painted with one coat of red-lead and then covered with fine cork and painted white. The cork does not cool so quickly as the steel, and so condensation does not take place so readily. This application is termed *cork cementing.*

(A very complete discussion of rusting, corrosion, and fouling in given in Professor Lewes's "Service Chemistry." See also a paper by Mr. Holzapfel, I.N.A., 1904.)

CHAPTER XII.

COALING.

THE rapid coaling of war vessels is a matter of considerable importance, and the present chapter is devoted to a brief consideration of the methods adopted for coaling in some recent large ships of the British Navy. The use of liquid fuel is still in the experimental stage; if it could be adopted, the getting on board and the storage would be a matter quite simple in comparison with what is necessary in the case of coal.

The subject of coaling divides naturally into three parts, viz.

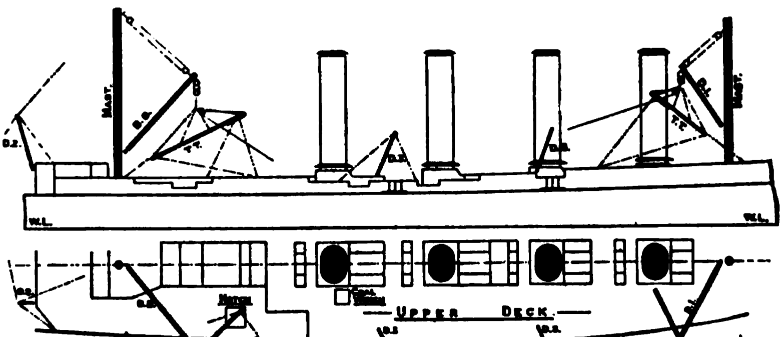


FIG. 112.—Coaling ship.

1. coaling ship, *i.e.* getting the coal on board from lighters, etc.; 2. getting the coal down into the several bunkers; and 3. getting coal from the bunkers to stokeholds. The two latter are specially difficult in war-ships, because of the large number of bunkers caused by the extensive system of watertight subdivision (see Figs. 52 and 53), and also by the presence of the armoured decks, which it is undesirable to pierce more than is absolutely necessary.

I. Coaling Ship.—Figs. 112 and 112A show the general

arrangements adopted in a large cruiser. These arrangements would, of course, be supplemented when coaling from a ship in which derricks, transporters, etc., are fitted for dealing with the coal.

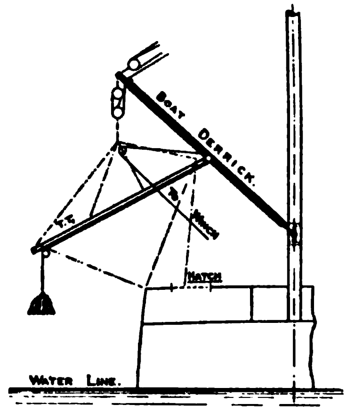


FIG. 112A.

There are two steam coal winches, one on each side of the upper deck, each having a central barrel and two side warping barrels. The coal is got on board by a Temperley transporter (two in some ships), which can be suspended either from the derrick on the foremast or from the main derrick on the mainmast. This transporter is 55 ft. long, and is fixed in position by means of guys. It has to be used in a slanting direction, so that the carrier will travel down by gravity when the coal-bags are empty. The transporter consists of an I beam, with a sheave on the upper end plumbing the upper deck; the lower end can be arranged to plumb the hold of the lighter.

The beam is provided with stops about 5 ft. apart, any one of which may be used to fix the load when it is desired to raise or lower. The lower end of the beam is fitted with a stop. The carrier itself runs on four rollers on the lower flange of the beam, and is composed of two side plates containing a number of cams. The purchase passes over a sheave on the carrier, and at its lower end has a heavy ball attached. Take the instant when the load is being lifted as Fig. 113. The double cam at the top is caught in the stop and the carrier is fixed. A further lift of the load, however, lifts up the pawl lever and catches the ball in the suspender hook as Fig. 114, and the weight is taken by the carrier. This movement of the levers, however, releases the double cam at the top away from the stop, and the whole carrier is then free to be hauled up the beam by hauling on the purchase. In order to fix the carrier to lower the load, it is hauled just past the desired stop as Figs. 115 and 116. The rope is then slightly slackened, and the toggle catches and turns the cam into the stop as Fig. 117. In this movement of the cam the carrier gets locked, and the ball gets released from the suspender hook and the weight

is free to be lowered as Fig. 113. When the empty coal-bags are lifted, the ball gets locked again by the suspender hook, and the carrier is loosened from the stop and is free to descend to the bottom of the beam by gravity. It will be noticed that a continuous motion past the stops is possible either up or down. If when going down the carrier is stopped just below the stop, then

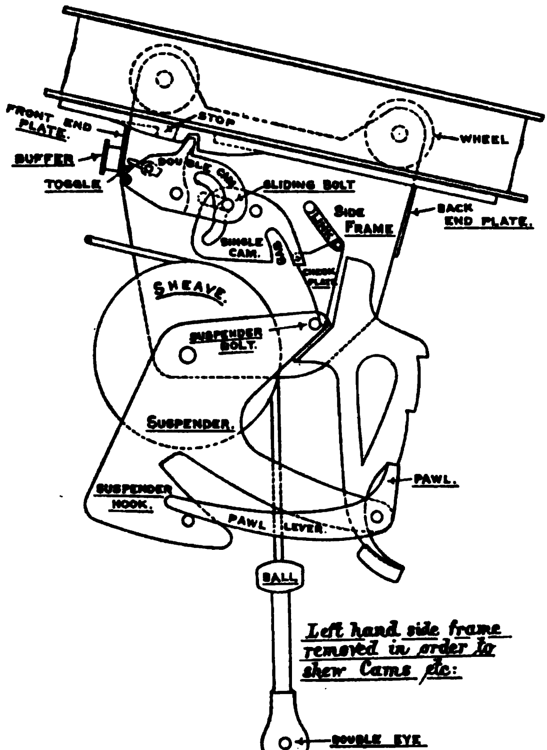


Fig. 113.

taken up just past it and the motion again reversed, the toggle shown so acts as to turn the cam into the stop.

The largest of these transporters can deal with 20 to 30 cwt. of coal at a time. There is only one rope to deal with, and the various operations are entirely under the control of the man working the coal winch.

Besides the above, portable derricks 20 ft. long are fitted at the gangway ports, with coaling platforms (9 ft. 6 in. × 3 ft. 0 in.)

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hung on to the side, on to which the coal-bags can be dropped and run in on to the upper deck. There are also provided portable

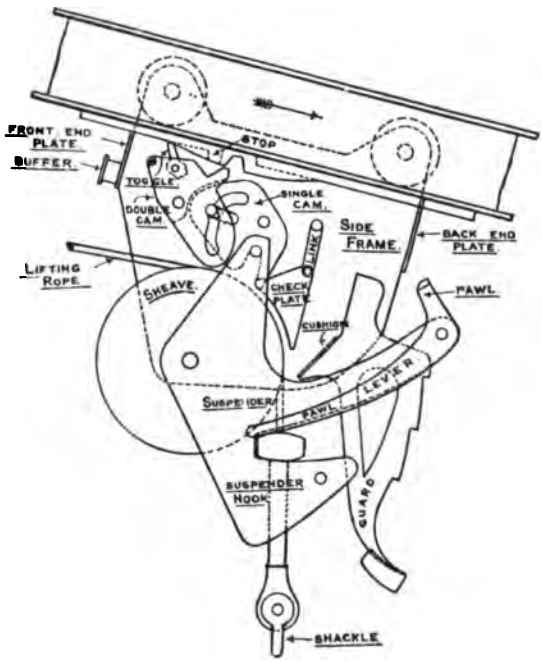


FIG. 114.

derricks on the after shelter bulkhead, which can be used for coaling purposes.

2. Getting Coal into the Bunkers.—It is not possible in a

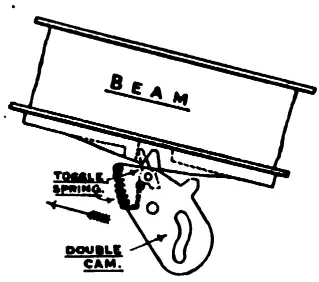


FIG. 115.

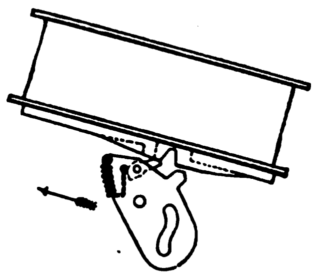


FIG. 116.

war-ship to have such large roomy bunkers as in merchant ships,

because of the necessity of providing complete watertight subdivision, and this renders the operation of getting coal into the bunkers, and from the bunkers to the stokeholds, correspondingly more difficult. In large merchant ships it is usual to have large cross bunkers, but this is not usually desirable in war-ships, because of the extra length thus entailed, and because a large proportion of the coal can be more economically arranged for

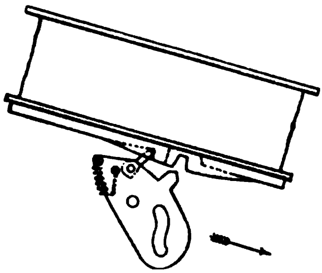


FIG. 117.

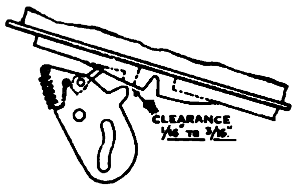


FIG. 118.

above the armour deck. There is also the fact that the coal thus stowed affords a valuable element of protection to the ship, this being of special importance in the case of deck-protected cruisers (Figs. 21 and 22) (see also Chapter XVII., in which the influence of coal on stability is dealt with). The lower bunkers are restricted in volume by the shape of the ship, and it is found that a large proportion of the total coal capacity is obtained above the pro-

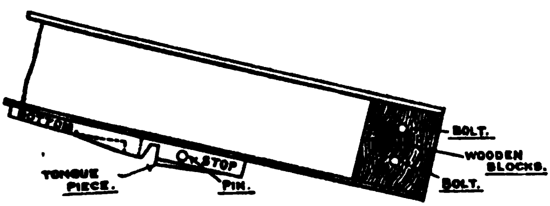


FIG. 119.

tective deck. The wings in a battle-ship (Fig. 12) are arranged for coal stowage when necessary, but these spaces are very much broken up by the deep framing, etc., there, and this renders the operation of getting the coal in and out of these spaces very difficult.

The ship selected to illustrate this part of the subject is the cruiser whose system of watertight subdivision is given in

Figs. 52 and 53. There are in this ship two series of upper bunkers, separated by a fore-and-aft watertight bulkhead, and one series of lower bunkers. The upper bunkers extend over the length of engine- and boiler-rooms; those over the engine-room have no fore-and-aft bulkhead, and the coal there may be looked upon as a reserve. The lower bunkers only extend over the length of boiler-rooms.

To coal the upper bunkers (Fig. 120) 20-in. scuttles are provided, as shown, in the main deck, one to each outer bunker and two to each inner bunker. Directly above these, on the upper deck, 18-in. scuttles are fitted. These pairs of scuttles are connected by coaling shoots, usually portable, in halves as shown; in some cases fixed shoots are provided. There is thus direct access from the upper deck to each of these upper bunkers. Escape scuttles are placed to each bunker in the main deck, close to a bulkhead underneath, with chains and footholds on the bulkhead.

To coal the lower bunkers there is a 33-in. coaling scuttle, as Fig. 139, in the middle deck. Above this, to the main deck, a rectangular trunk is fitted. Above this, in the main deck, a 20-in. scuttle is fitted, with an 18-in. scuttle, immediately over, in the upper deck. These latter are connected by a portable shoot in the ordinary way. There is thus direct access for coal from the upper deck to the lower bunkers. Escape doors are provided in the side of the ammunition passage, with ladders and hand chains as necessary for getting out.

3. Getting Coal from the Bunkers to the Stokeholds.—Doors are provided from each stokehold into the lower bunkers. It is necessary, therefore, to get the coal from the upper to the lower bunkers when the latter are getting empty. The rectangular trunks are provided with doors on each side as shown. The outer side, which is a watertight bulkhead, has a sliding watertight door. The other doors are not watertight. They are made in two pieces, so that the lower half hinges up and is secured to the latter, and then both are hinged up clear of the opening and made fast. With these doors all open, and the 33-in. scuttle open, coal can be trimmed from either of the upper bunkers to the lower bunkers. An additional scuttle is provided at the side of each upper outer bunker, and a shoot is taken down as shown, so that coal may be passed into the lower bunkers through the watertight door on the wing bulkhead.

A set of rails is fitted throughout the upper bunkers in order to

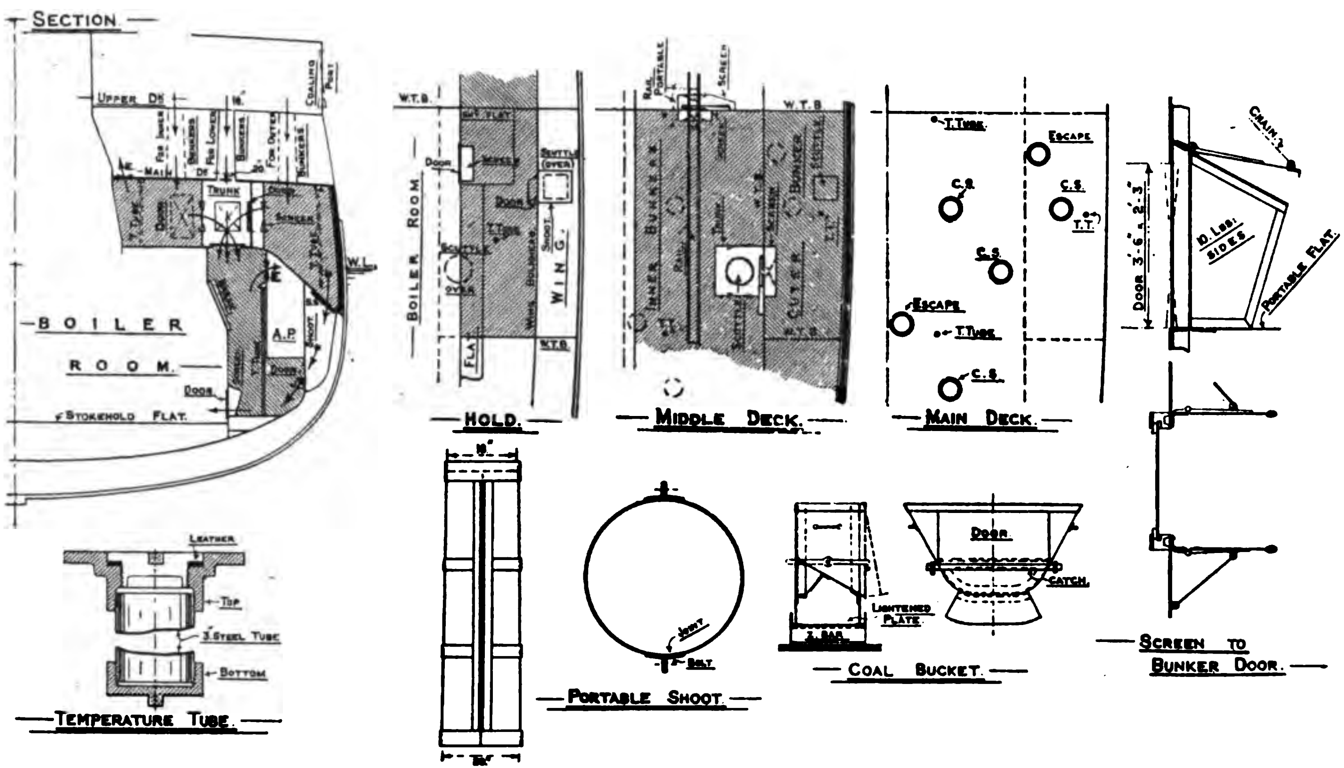


FIG. 120.—Coaling arrangements.

enable a coal-bucket to be run along to bring the coal from any upper bunker to any desired lower bunker. The coal-bucket is shown in Fig. 120, and has one side hinged and a sloping bottom. The bucket would be brought up to the trunk and the catch released, and the contents dropped into the lower bunkers. The rail has to be made portable in way of the watertight bulkheads to allow the door to close. Most of these are horizontal sliding doors worked from the main deck, but in the after bunkers they are ordinary hinged doors.

In recent cruisers the wing bulkhead and the divisional bulkhead in the upper bunkers has been dispensed with (see Fig. 23). This considerably increases the coal capacity, and makes the operations of coaling very much simpler.

Care is taken in all coal-bunkers to screen the watertight doors from the pressure of the coal, so that the doors may close and open even if the bunker is full of coal. A specimen screen is shown in Fig. 120. Two side plates support a sloping top plate, and these keep the coal from the door. When the door is free from coal the side plates can be hinged back, and the top plate hinged up out of the way. Screens are also provided, as shown, to the doors in the upper bunkers.

We have seen in Chapter X. the necessity for the efficient ventilation of coal-bunkers and the provision made for the same. It is also necessary to provide means for ascertaining what the temperature is in the body of the coal. For this purpose temperature tubes are placed as shown. For the lower bunkers the temperature tubes are approached from the ammunition passage, for the upper bunkers from deck plates on the main deck. The temperature has to be noted at frequent intervals.

CHAPTER XIII.

ARMOUR AND DECK PROTECTION.

THE three methods of attack which a war-ship may have to withstand are ramming, torpedo- or mine-attack, and gun-fire. The two former would cause damage principally at and below the waterline, and the only protection that can be afforded is the extensive watertight subdivision. For protection against gun-fire, armour plates are provided over as large a portion of the ship as possible. These armour plates are, in most cases, backed up by coal and thick decks. The parts not protected by armour are minutely subdivided in the neighbourhood of the waterline in order to localize damage as far as possible. Thick decks, both above and below water, are also largely employed for purposes of protection.

A large proportion of the weight set aside for protection in modern ships is necessary for the protection of the armament. Thus in a recent battle-ship the total weight provided for protection was 4335 tons, and this is divided as follows:—

Armour and deck protection to hull for the preservation of buoyancy and stability, 2875 tons.

Protection to armament, as barbettes and casemates (not including gun shields), 1460 tons.

This shows that, in this case, about one-third the available weight was devoted to the protection of the armament and two-thirds to the ship.

It is proposed to trace briefly the history of the subject from the *Warrior* until the present time, taking typical ships. The dates given refer to the year in which the ships were laid down.

"Warrior."—The first large vessel provided with armour protection in this country was the *Warrior* (1859). This ship was built of iron, and was 380 ft. long, and 9200 tons displacement. The armour was 4½ in. thick, and extended, as shown in Fig 121, for a length of 218 ft. and a depth of 22 ft. This 4½-in.

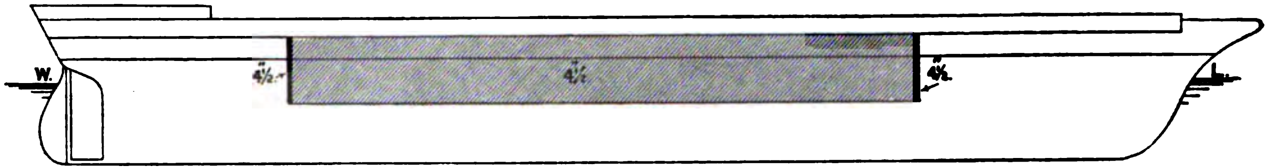


Fig. 121.—H.M.S. *Warrior*.

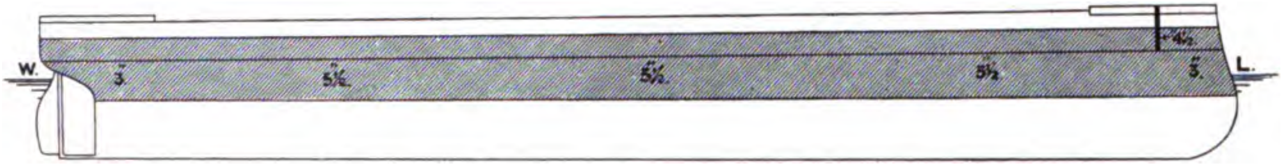


Fig. 122.—H.M.S. *Minotaur*.

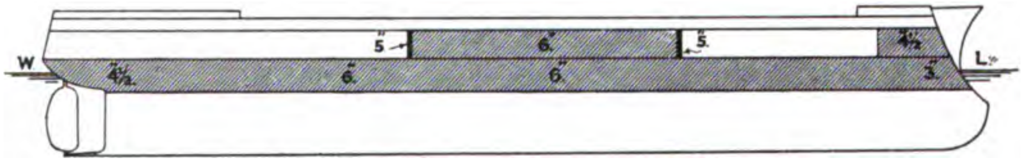


Fig. 123.—H.M.S. *Bellerophon*.

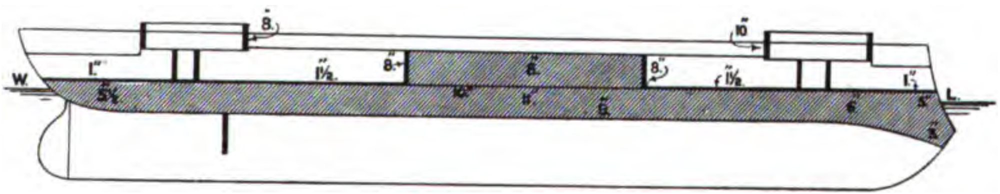


Fig. 124.—H.M.S. *Temeraire*.

armour was the thickest that could then be produced, and experiments carried out at the time showed that it was sufficient to withstand the guns then in use. The armour was of wrought iron, and this material was used for protection right up to the *Inflexible* (1874). Owing, however, to the continuous development of gun power the thickness and disposition of the armour in subsequent ships underwent considerable modifications. One serious disadvantage in the *Warrior* was the unprotected state of the ends and the rudder-head. This was remedied in subsequent ships.

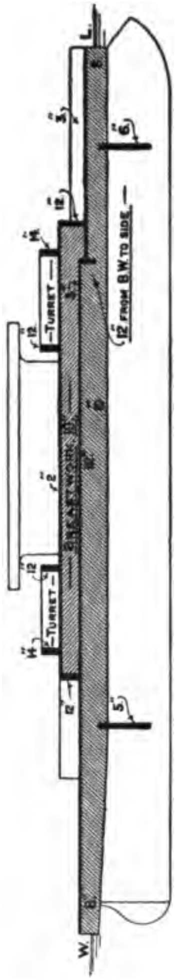


FIG. 125.—H.M.S. *Devastation.*

"**Minotaur.**"—This ship (1861) was larger than the *Warrior*, being 400 ft. long and 10,690 tons, the extra size being necessitated by the increase of the thickness of armour to 5½ in. amidships (Fig. 122); and the provision of 3-in. armour at the ends.

"**Bellerophon**" (1864).—The next step was the production of a shorter and handier ship, the *Bellerophon*, which was 300 ft. long and 7550 tons. There was a complete belt at the waterline (Fig. 123), 6 in. thick amidships, tapering at the ends as shown. Above the belt for a length of 94 ft. amidships there was an armoured battery of 6-in. armour.

"**Temeraire.**"—The above type of ship with a complete belt and a battery amidships continued until the *Temeraire* (1873), in which ship the belt was 11 in. maximum thickness, with a battery 8 in. thick (Fig. 124). In this ship we also find 8-in. and 10-in. redoubts towards the ends for mounting the heavy guns. The *Temeraire* was 285 ft. long and 8540 tons.

"**Devastation**" (1869).—This was a special type of ship in which sails were abandoned and twin screws adopted. She had low freeboard (Figs. 125 and 125A), and was 285 ft. long and 9320 tons. The belt, 12 in. thick amidships, was continued to the ends 8 in. thick. Above this belt there was a central breastwork 153 ft. long, 10 in. thick along the sides, and

12 in. round the turrets. These turrets were 14 in. maximum thickness.

During the above period (1859 to 1873) we have noticed that, owing to successive improvements in guns, continuous increase had to be given to the thickness of armour protecting the ships, and a stage was at length reached when it was found impossible to cover any large area of a ship's side with armour thick enough to resist the fire that could be brought to bear against it. The next development accordingly consisted in only attempting to protect a portion of the ship with very thick armour, and to depend on deck and other protection for the remainder. The principle acted upon was that it would be better to efficiently protect the midship portion of the ship in way of machinery and heavy guns, than to cover a large area with thinner armour that could not keep out the enemy's fire. This principle was carried to its extremest limit in the *Inflexible*.

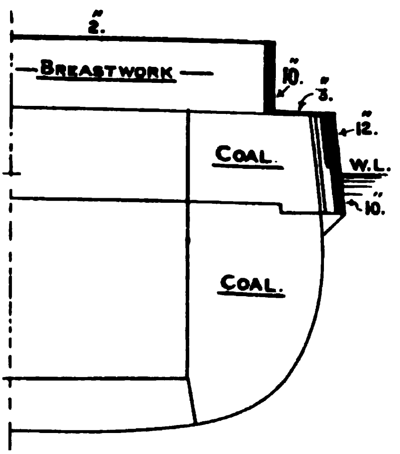


FIG. 125A.—H.M.S. *Devastation*.

"Inflexible" (1874).—This ship was 320 ft. long and 11,880 tons. The armoured citadel (Fig. 126) was 110 ft. long, with a maximum thickness of 24 in., the armour being in two thicknesses. Forward and aft of this citadel the magazines, etc., were protected by under-water decks, 3 in. thick, and some protection was afforded by the stowage of coal and cables on these decks, with cork packing at the sides and cofferdams, as shown.

In this ship the hull armour was still of iron, but the outer thickness of the turret armour, 9 in., was "*compound*," or *steel faced*.

Figure of Merit of Armour.—The resistance of armour to penetration is compared with the penetration of wrought iron as found by some empirical formula. There are a number of these formulæ, the one used in England being that obtained by Captain Tresidder (see "Gunnery Manual"). He found that penetration of wrought iron could be represented by the formula—

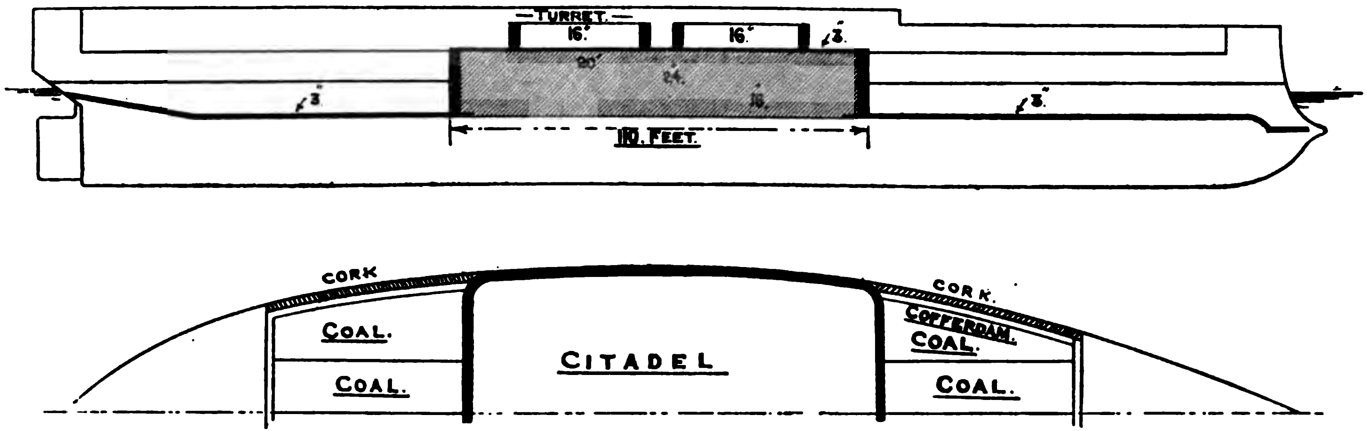


FIG. 126.—H.M.S. *Inflexible.*

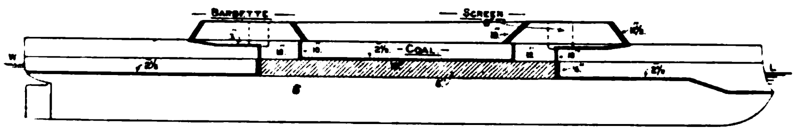
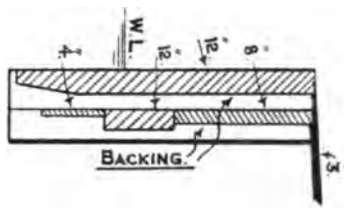


FIG. 127.—Armour, etc., Admiral class.



—SECTION OF SIDE.—

FIG. 126A.

$$t^2 = \frac{WV^3}{D} \times \frac{1}{693,500,000} \qquad (\log 693,500,000 = 8{\cdot}841)$$

where t = thickness of wrought iron in inches
V = striking velocity in feet per second
D = calibre of shot in inches
W = weight of shot in pounds.

The ratio of the thickness of wrought iron as found by this formula to that of the armour plate tried is termed the *figure of merit.*

EXAMPLE.—A 6-in. armour plate is attacked by a 6-in. 100-lb. Holtzer armour-piercing shot with 2177 ft. per second striking velocity. The plate resists penetration, find the figure of merit—

we have $t^2 = \dfrac{100 \times (2177)^3}{6} \times \dfrac{1}{693,500,000}$ for wrought iron.

By using logarithms t is found to be 15·7 in., so that the *figure of merit* is 2·6.

Compound Armour.—A compound armour plate consists of a wrought-iron plate artificially attached to a steel face of about half its own thickness, the result being a plate with *hardness of steel* on the face by which projectiles are broken up, and *toughness of wrought iron* at the back which prevents cracking taking place. As at first made this armour had a figure of merit of about $1\frac{1}{2}$, but improvements in manufacture brought up the ratio to 1·7. In France all steel plates were used, but experiments showed that this material was lacking in toughness, and trouble was caused by the spontaneous cracking of these plates. Compound plates were used in ships after the *Inflexible* up to and including the *Royal Sovereign* (1889).

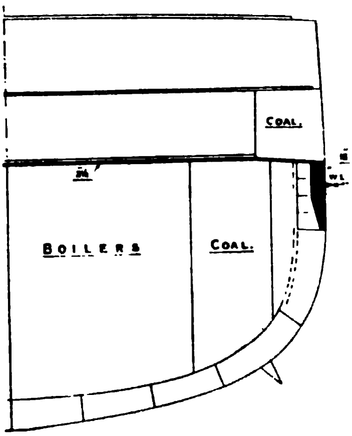


FIG. 127A.—Section Admiral class.

Admiral Class.—The next distinctive type of ship was that known as the Admiral class (1880). The various ships of the class differ somewhat in detail, but their main features are the same. The *Collingwood*, 325 ft. long and 9500 tons (Figs. 127 and 127A), may

be taken as a typical ship of the class. The armour belt was 18 in. thick amidships, 7 ft. 6 in. broad, and it extended for a length of 140 ft., or about seven-sixteenths the length of the ship with athwartship bulkheads. No protection save coal was provided above this armour belt. Forward and aft were two sloping barbettes of 11½ in., with trunks coming down within the limits of the armour belt as shown. The floor of these barbettes was 3 in. The ends of the ship were unprotected by vertical armour, but underwater decks were worked 2½ in. in thickness. The top of the belt was covered in with a level deck 2½ in. thick.

While this class of ship was building, quick-firing guns and high explosive shell were being developed, and although only 6-pounder shell were projected, yet it was felt that the large areas of unprotected side in these ships was an element of serious danger. The influence of this factor is seen in the *Nile* and *Trafalgar* (1886), and in the next main type of ship, the *Royal Sovereign* class.

"Royal Sovereign" (1889).—This ship was 380 ft. long and 14,150 tons displacement. There were eight of the class built,

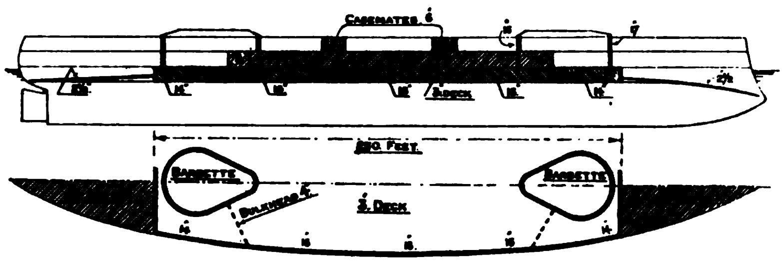


FIG. 128.—Armour, etc., H.M.S. *Royal Sovereign*.

seven with barbette mountings for the heavy guns (*Royal Sovereign, Ramillies, Repulse, Royal Oak, Resolution, Revenge, Empress of India*), and one with turrets (*Hood*). The former ships were able to obtain a greater height of big guns above water than the *Hood* on account of the weight involved in the turrets.

In these ships the waterline belt was 8½ ft. broad, with a maximum thickness of 18 in., covered in with a horizontal deck 3 in. thick (see Figs. 11 and 128). The length of the belt was 250 ft., or two-thirds the length of ship. Bulkheads, 16 in. and 14 in., closed in the belt at the forward and after ends respectively. The one-sixth of the length at the ends had underwater decks 2½ in. thick. Above the 18-in. belt, from middle to main decks,

4-in. armour is worked, with an upper coal-bunker behind. This armour was of nickel steel, and was fitted in order to determine the explosion of shell filled with large bursting charges of high explosive outside the ship, and to prevent the free perforation of the side above the belt by the smaller nature of quick-firing guns.

The barbettes mounting the 13½-in. guns were pear-shaped and of very substantial construction, the maximum thickness being 17 in., and the armour extending right down to the middle deck. The armour for this class was compound, except the 4-in. side, which was of nickel steel, and the main deck casemates of steel.

Harveyed Armour.—The next large group of ships were those of *Majestic* class, nine in number. In these ships armour made by the Harvey process was adopted. This armour is estimated to have a figure of merit of about 2, as against 1·7 for compound armour. The development of armour-piercing projectiles of forged steel was the cause of this improvement in armour manufacture. In the Harvey process an all steel plate is used, and the face is *cemented*, *i.e.* animal charcoal is placed next the face of the plate (two plates being usually dealt with together, face to face), and the whole is covered in with bricks and run into a gas furnace, where it remains two to three weeks, seven days or so being allowed for cooling. In this way the proportion of carbon on the face is increased, and the front is then capable of being hardened. The plate is first cemented as above, and then bent to the required shape and all necessary holes made in the surface. It is then heated and the face douched with cold water, which makes the front of the plate exceedingly hard. We thus have a compound plate, but the junction between the hard face and the tough back is much more perfect than in the compound plates. The object to be attained was a steel plate, without welds, having such a proportion of carbon in the surface that water cooling would produce a very hard face. As the thickness of the hard steel is practically constant for all thicknesses of plate, it follows that thin plates obtain relatively higher values of the figure of merit than thicker plates. That is, a 12-in. plate is not twice as good as a 6-in. plate. For Harveyed armour a figure of merit of 2 may be taken, for thinner plates (6 in.) 2¼.

"Majestic" (1894).—There are nine ships in this class (*Majestic, Magnificent, Mars, Jupiter, Hannibal, Illustrious, Victorious, Prince George, Cæsar*). They are vessels of high freeboard, and are 390 ft. long and 14,900 tons (Figs. 129 and 130). A new departure

was taken in these ships in the disposition of the armour. The belt was 9 in. only, but it was carried up to the main deck for a total depth of 15 ft. The length of the citadel was 250 ft., or about two-thirds the length. The protective deck between barbettes was run level across at the middle line, but sloped down to

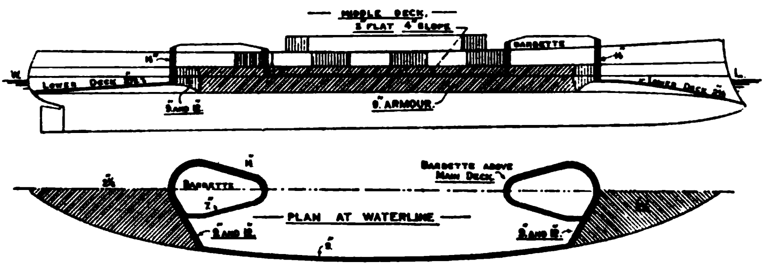


Fig. 129.—Armour, etc., H.M.S. *Majestic*.

the bottom of the armour at the side, as shown in Fig. 130. This deck was 3 in. on the flat and 4 in. on the slope. In this arrangement of armour the influence of the larger nature of quick-firing

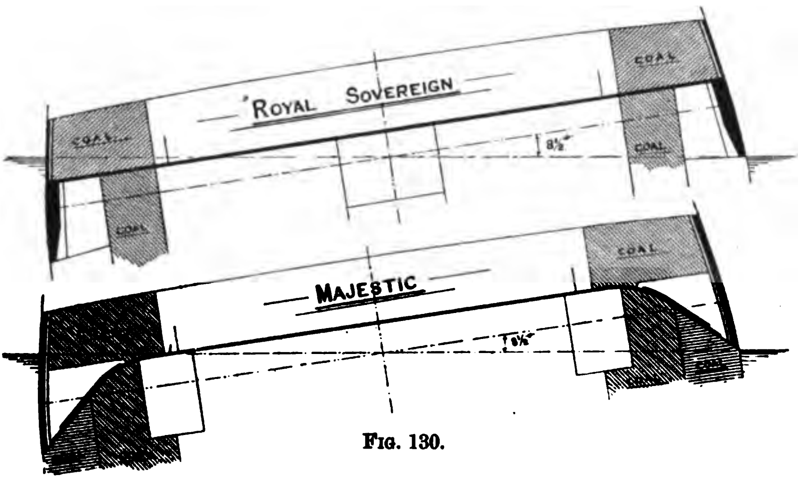


Fig. 130.

guns is seen. The protection at the waterline is not sufficient to keep out the heaviest projectiles. It is, however, backed up by the 4-in. sloping deck (which is about equivalent to 5 in.

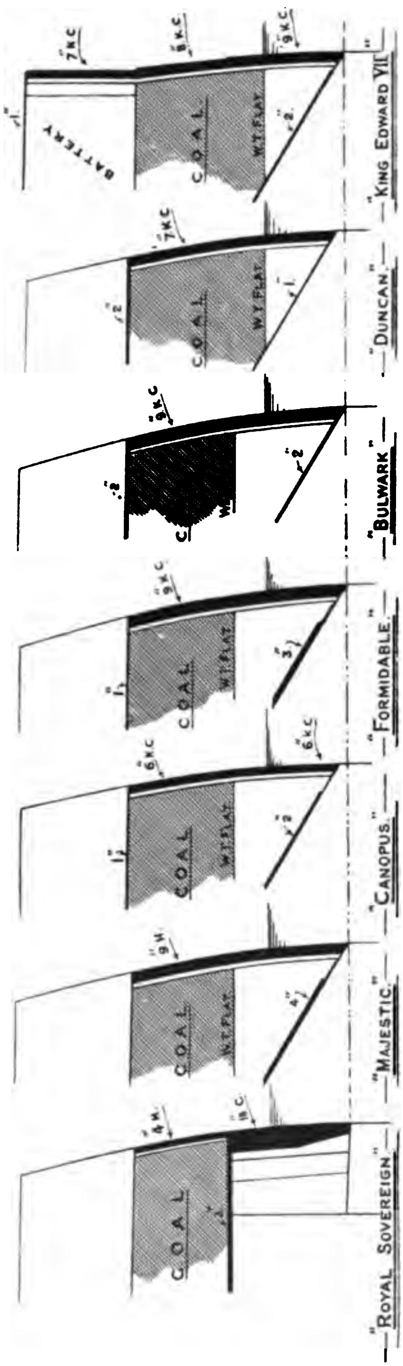


FIG. 131.—Sections at side of battle-ships. C., Compound; N., Nickel; H., Harvey; K.C., Krupp cemented.

horizontally), so that before the vitals of the ship are reached a penetration equivalent to about 23 in. of wrought iron is necessary. Although this does not compare well with the 30 in., say, in the *Royal Sovereign*, yet it is certain that a ship will be most difficult to hit at the waterline, and it was considered better to give up absolute protection at the waterline in order to obtain a larger area of good protection, because of the development of the large quick-firing guns. The ends of the ship are not protected by vertical armour, but have underwater decks 2½ in. thick. In some of the later ships of this class the barbettes are circular, and not pear shaped as Fig. 129, owing to a change of the type of gun mounting.

A point worth noting in connection with a narrow belt (as in Fig. 132) is the fact that when deeply loaded, or damaged so that sinkage takes place, the top of the belt may possibly be at or below the waterline, and the advantage of a thick belt is then lost. Another point is that ships increase their draught as time goes on (about an inch yearly), owing to alterations and additions, so that the value of the belt gets less and less during the course of years, unless steps are taken to reduce the draught.

Krupp Armour.—The Harvey process of manufacturing armour was soon superseded by the Krupp process. The steel for this process has a high tensile strength, approaching 50 tons per square inch, and contains small proportions of nickel, chromium, and manganese. Plates above 4 in. are cemented, and are termed K.C., or Krupp cemented. Plates 4 in. and below are not cemented,

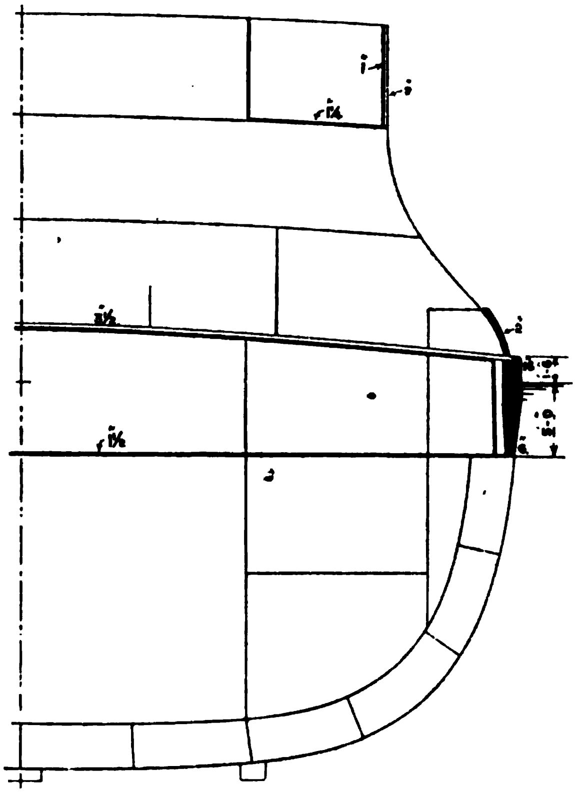


FIG. 132.

these are termed K.N.C., or Krupp non-cemented. For K.C. plates the cementation is carried out in a similar manner to that of the Harvey process, but in the final face hardening the plate is not heated bodily as in the Harvey process, but the heat is graduated from the face to the back. After heating the face is placed under the cold water douche.

For K.N.C. plates the composition of the steel is similar, but the plate is water cooled without previous cementation. These plates are about equal in resisting power to Harvey plates, *i.e.* have a figure of merit of about $2\frac{1}{4}$. K.C. plates have a figure of merit about 2·3 to 2·5. In some few cases a higher figure of merit, approaching 3, has been obtained.

(An exhaustive summary of the armour question year by year is given in Lord Brassey's *Naval Annual.* See also the supplements to Captain Orde-Browne's "Armour and its Attack by Artillery.")

Armour Bulkhead.—In the case of a thwartship armour bulkhead to a ship with a sloping protective deck as *Majestic* and following ships, it is necessary to pay attention to the thickness of this bulkhead where the middle deck does not come behind. Fig. 133 shows the after armour bulkhead of a battle-ship with 7-in. side armour. The armour protecting the barbette gun

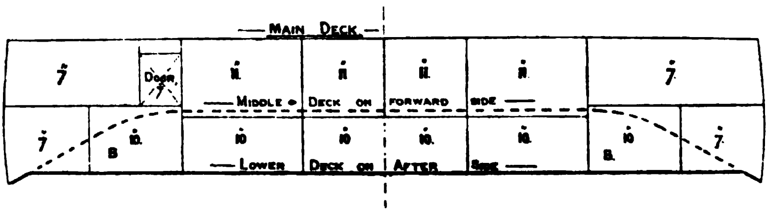


FIG. 133.—After armour bulkhead.

mountings is 11 in. and 10 in. At the points marked B there is no deck behind the armour, and a shot striking there would have direct access to the vitals of the ship. Accordingly, the lower bulkhead plates shown are made 10 in., considerably thicker than the ordinary side armour.

"Canopus" Class (1897).—The vessels of this class are six in number (*Canopus, Albion, Glory, Ocean, Goliath, Vengeance*). They are 390 ft. long and 12,950 tons. The belt armour is 6 in. only, and there are two protective decks, the lower one being 2 in. and the upper one, the main deck, being 1 in. The citadel is 227 ft. long, and the forward end of the ship is protected with 2 in., worked on a double thickness of plating. The belt is closed in with armour bulkheads at each end, and the barbettes are circular, having a maximum thickness of 12 in. The lower decks, forward and aft, are 2 in. A section of the side amidships of this class is given in Fig. 131.

"Formidable" Class.—The next type of ship laid down (1898) was the *Formidable* class (*Formidable, Implacable, Irresistible*). These ships are 400 ft. long and 15,000 tons. They are protected similarly to the *Majestic*, with 9-in. armour, but the main deck is made 1 in., and the middle deck 2 in. on the flat and 3 in. on the slopes (see Fig. 131 for section at side). The lower decks at ends are 2 in. forward, 2½ in. aft, and the side forward has 2 in., worked on a double thickness of plating. The side plating aft is increased to a total thickness of 1½ in. The barbettes have a maximum thickness of 12 in.

"Bulwark."—The five later ships of *Formidable* class (*Bulwark, London, Venerable, Queen, Prince of Wales*) (1899) are of the same dimensions and displacement as the *Formidable*, but the armour is arranged differently at the forward end. The forward armour bulkhead is dispensed with, and the side armour is carried forward from the 9 in. side in steps of 7 in., 5 in., and 3 in., and right forward we have 2 in. worked on a double thickness of skin plating. The main deck is 2 in. over the citadel, and is worked to the stem, where it is 1 in. thick. The middle deck in way of citadel is 1 in. on the flat, 2 in. on the slopes. The lower deck forward is 1 in., and where it slopes to meet the middle deck before the forward barbette it is 2 in. The lower deck aft is 2½ in. The side plating aft is increased to a total thickness of 1½ in. The whole arrangement is similar to that of the *Duncan* (Fig. 134).

"Duncan" Class (1899).—The six vessels of this class (*Duncan, Cornwallis, Exmouth, Russell, Albemarle, Montagu*) are 405 ft. long and 14,000 tons. They were specially designed for high speed, viz.—

19 knots on	8 hours' trial,		18,000 I.H.P.
18 „	30 „	„	13,500 „

In consequence of this, these ships had less displacement and a finer form than *Formidable*, in order to avoid excessive I.H.P., and the armament being the same as in that ship, the weight available for protection would only allow a 7-in. belt to be worked. This belt is reduced in thickness to the forward end, as shown in Fig. 134. Aft the plating is doubled, and on the flush surface thus obtained 20-lb. (½ in.) nickel steel is worked. The decks are as follows:—

Main deck: 2 in. between barbettes to 1 in. right forward.
Middle deck: 1 in. uniform between barbettes.

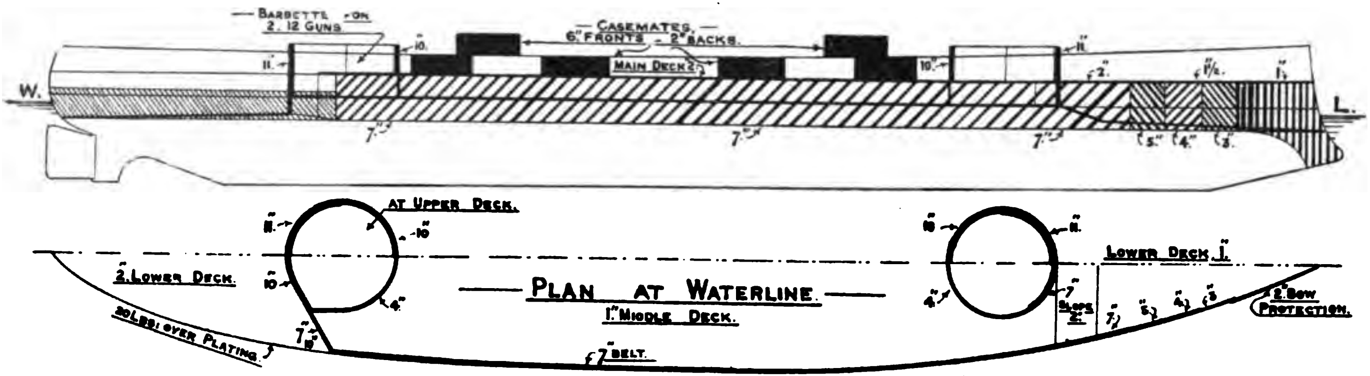


Fig. 134.—Armour H.M.S. *Duncan*.

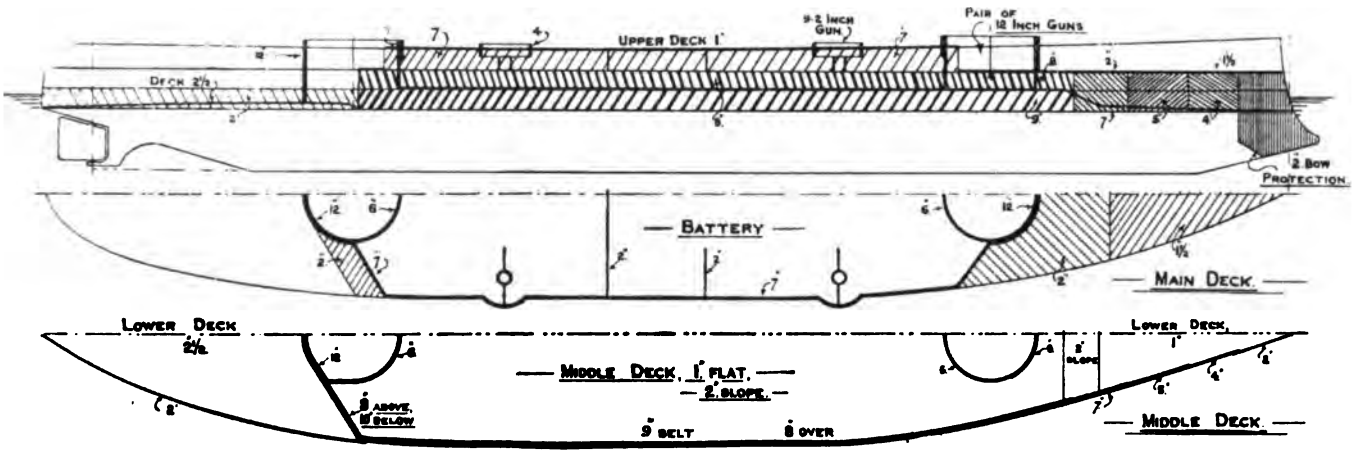


Fig. 135.—Armour protection, etc., H.M.S. *King Edward VII.*

Lower deck: 1 in. forward, 2 in. aft; slope to middle deck forward, 2 in.

In all the above ships back to the *Majestic* the secondary armament of 6-in. guns are carried in isolated casemates having 6-in. fronts and 2-in. backs.

"King Edward VII." (1902).—Of this class there are eight building at the present time, viz. *King Edward VII.*, *Commonwealth, Hindustan, Dominion, Hibernia, Britannia, Africa, New Zealand.*

The particulars are: length, 425 ft.; breadth, 78 ft.; draught, 26 ft. 9 in.; displacement, 16,350 tons; I.H.P., 18,000; speed, 18½ knots; armament, 4 12-in., 4 9·2-in., 10 6-in. In these ships the casemate system of protecting the secondary armament has been adandoned, and a battery, 7 in. thick, has been worked between the main and upper decks to take the 6-in. guns (see Figs. 13 and 135). The battery is covered in with 1-in. plating at the top, and this battery also performs the function of protecting the funnel casings, etc., to the upper deck.

The armour belt is 9 in. at the waterline, 8 in. above to main deck. This belt is carried in reduced thicknesses to the bow, as shown in Fig. 135. A bulkhead is worked at the after end of citadel as usual. In this ship also the after end is protected by 2 in. worked on doubled plating. The decks are as follows:—

Upper deck over battery, 1 in.
Main deck, forward of battery, 2 to 1½ in.
Middle deck, 1 in. flat, 2 in. slope.
Lower deck, forward, 1 in., aft 2½ in.

The barbettes for the 12-in. guns have a maximum thickness of 12 in., reductions being possible where the battery, etc., would also have to be pierced before reaching the barbette. The shallow barbettes for the 9·2-in guns are 4 in. thick.

Fig. 131 gives an interesting comparison of the armoured sides of British battle-ships from *Royal Sovereign* to *King Edward VII.*

Protection of Cruisers.—The essential quality to be obtained in cruisers is high speed and large coal supply, and as much protection is given as the limited weight available will allow. In the belted cruisers of the *Orlando* class, protection was obtained by a narrow belt, 5½ in. wide, 10 in. thick, extending over about two-thirds the length. There was a level deck 2 in. over the belt, and decks at the ends 3 in. thick. Some disadvantages of a narrow

belt have already been noticed, and since the *Orlando* until the time of *Cressy* class (1897) large cruisers were "protected,"[1] *i.e.* they had the vitals covered in with a thick deck, and this thick deck, in conjunction with the coal stowed above it, was depended upon to give protection. We shall see in Chapter XVII. how important this coal is, not only in offering direct resistance to penetration, but in preserving stability when the side is riddled.

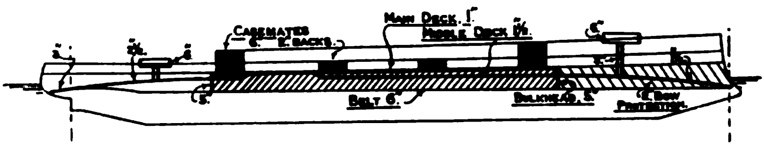


FIG. 186.—Armour, etc., H.M.S. *Cressy*.

Examples of large protected cruisers are given in Figs. 21 and 22, and the smaller cruisers of the second and third classes are still constructed on this system (Figs. 24, 26, 27). The side bunkers are also well subdivided by watertight bulkheads in order to localize any damage at the waterline (see Fig. 54).

"**Cressy**" (1897).—The ships of *Cressy* class represent a distinct

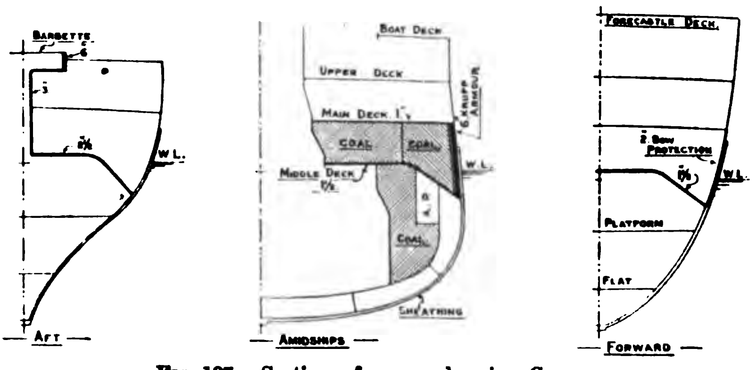


FIG. 187.—Sections of armoured cruiser *Cressy*.

departure in the design of large cruisers. There were six ships of this class, all being sheathed with teak and copper (*Cressy, Aboukir, Hogue, Sutlej, Bacchante, Euryalus*), 440 ft. long, 12,000 tons. Advantage was taken of the improved quality of armour obtained by the Krupp process to armour the side for about half

[1] *Edgar* class; *Blake, Blenheim, Powerful, Terrible, Diadem* class.

length and a depth of 11½ ft. with 6-in. armour, closing in the ends with bulkheads 5 in. thick (Figs. 136, 137). Two-in. protective plating was worked to the bow directly on the ship's plating. The middle deck is 1½ in., and main deck 1 in. in way of armoured side. The lower deck forward is 1½ in., and aft 2½ in. In a cruiser, armoured in this way, the height of the middle deck above water can be made less than in a protected cruiser, because of the presence of the armoured side (compare Figs. 22 and 23).

"Drake" (1899).—The four cruisers of *Drake* class (*Drake, King Alfred, Leviathan, Good Hope*) are unsheathed, and are 500 ft. long, 14,100 tons, and with 30,000 I.H.P. were designed for the high speed of 23 knots. The protection is similar to the *Cressy*, with a 6-in. armour belt and 2-in. protection forward.

"Monmouth" (1899).—The first cruisers of this class (*Monmouth, Bedford, Kent, Essex, Suffolk, Cornwall, Cumberland, Berwick, Lancaster, Donegal*) are 440 ft. and 9800 tons, 22,000 I.H.P., 23 knots. They are armoured with a 4-in. belt, with middle deck ¾ in., and main deck 1¼ in., as Fig. 23. The later cruisers have a 6-in. belt, and are somewhat larger, viz. 450 ft. long and 10,700 tons, 21,000 I.H.P., 22¼ knots (*Devonshire, Antrim, Roxburgh, Carnarvon, Hampshire, Argyll*).

"Duke of Edinburgh" (1903).—The first class cruisers (*Duke of Edinburgh, Black Prince*), are 480 ft. long and 13,550 tons, 23,500 I.H.P., 22⅓ knots. They are similar to *King Edward VII.* in having an armoured battery (Fig. 138). This battery contains the ten 6-in. guns of the secondary armament, and is 6 in. thick. The main armament consists of six 9·2-in. guns on the upper deck and forecastle in barbettes. The belt is worked over the whole length, 6 in. thick amidships, 4 in. forward, and 3 ft. aft. The upper deck over battery is 1 in., the main deck forward and aft is 1 in., the lower protective deck is ¾ in., being thickened up to 3 in. over the steering gear. The later ships of the class—*Warrior, Achilles, Natal, Cochrane*—are being considerably modified in respect to armament.

Second and Third Class Cruisers.—As mentioned above, these cruisers are still being built on the deck-protected system, Fig. 24 showing a typical sheathed second class cruiser, Fig. 27 showing a third class cruiser. The new class of *Scouts* now building are also protected in the same way.

Sloops.—In these vessels (Fig. 29) no attempt is made at protection, except that in the bunkers a sloping or level partition

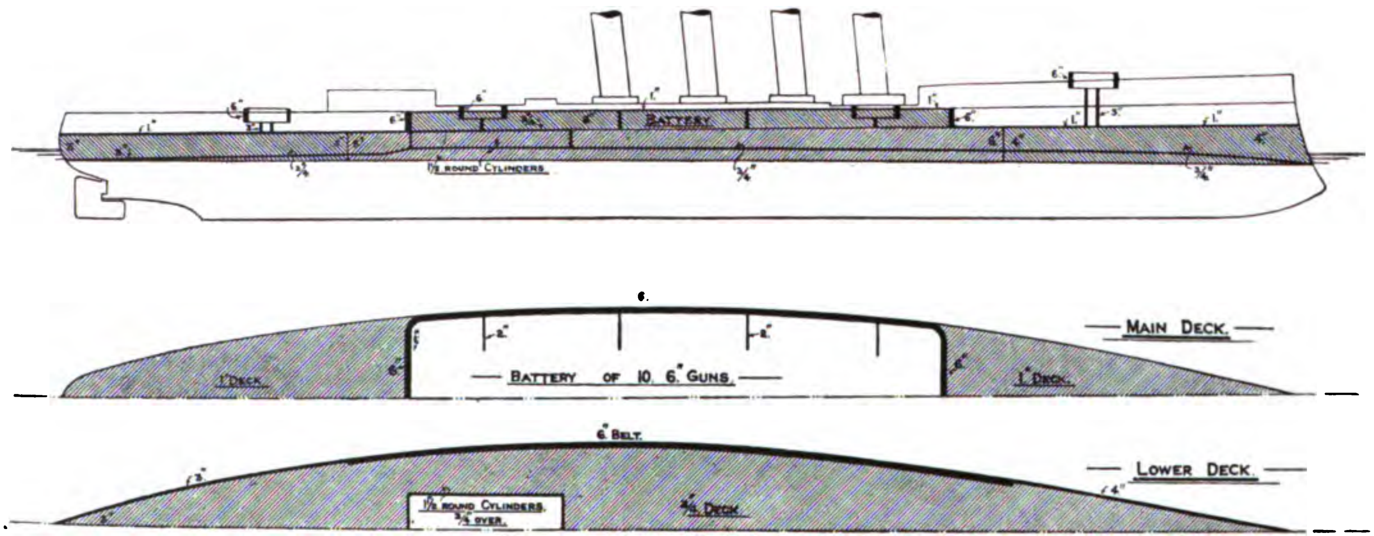


Fig. 188.—Armour, etc., H.M.S. *Duke of Edinburgh*.

is placed near the waterline, so that coal could be retained above, if desired, to obtain the protection which the coal affords.

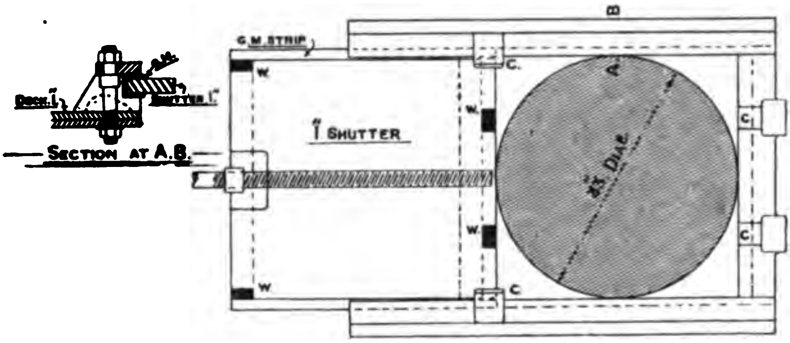


FIG. 139.—Sliding scuttle to armour deck.

Armour Scuttles and Gratings.—In protective decks there are many openings for access, etc., which it would be necessary to

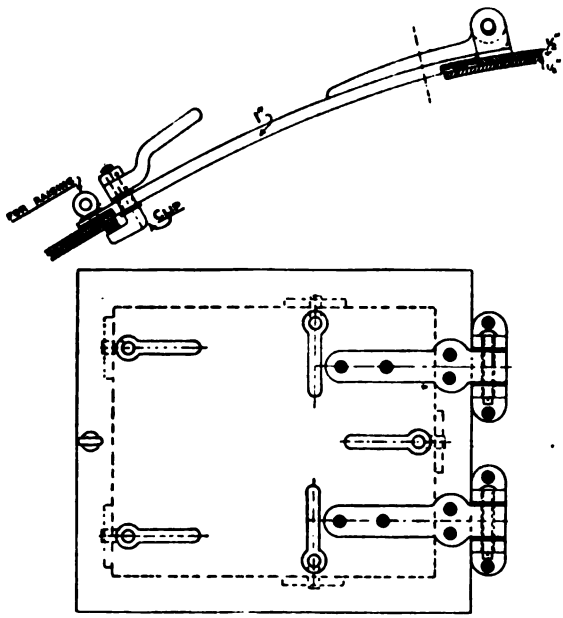


FIG. 140.

have closed in action to preserve the protection. These openings are usually closed by hinged covers having the same thickness as

the deck. In coal-bunkers, where access is not possible, the holes are frequently covered by a sliding scuttle. A sliding scuttle is shown in Fig. 139. The plate works like a vertical sliding door, except that the grooves are parallel and not tapered. The wedges W jam under the clips C when the scuttle is closed. The gearing is worked from a convenient position outside the bunker, and the scuttle and gearing is covered in, so that it can be opened when the bunker above is full of coal. The usual scuttle in the protective deck is hinged, as Fig. 140. In such a scuttle, however, fitted in a coal-bunker, access must be provided by a trunk so that the clips may be got at to secure the cover. Scuttles of a

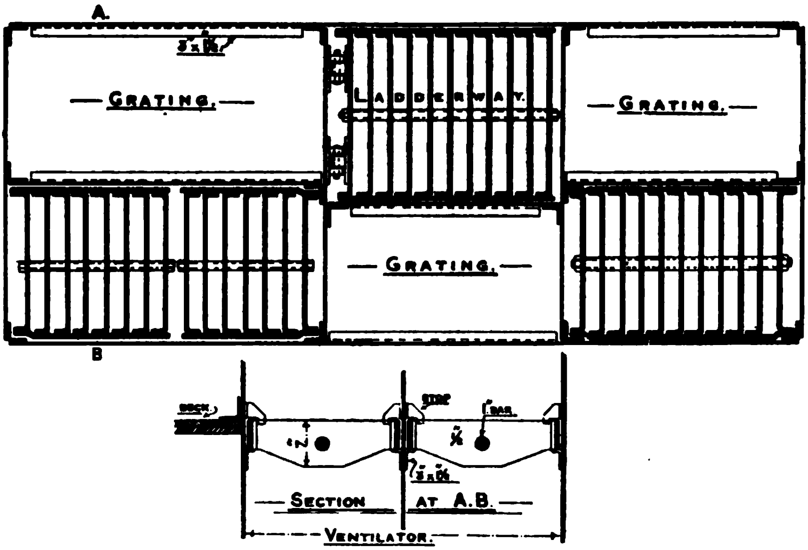


FIG. 141.—Armour gratings.

similar nature are fitted to the torpedo hatches, etc. Scuttles which are provided for escape are fitted with a balance weight so that they may lift easily.

There are, however, a number of openings in these decks which must be open even when the ship is in action. Such are the funnel and ventilator openings, and in these are fitted *armour gratings.* Fig. 141 shows a specimen set of gratings in a ventilating shaft. The space is divided into a number of rectangular spaces by girders, and into each of these spaces one or more gratings is fitted, resting on angles as shown in the sketch.

For a 2-in. deck the gratings are 7 in. deep, ½ in. thick, and the clear space between the bars is 2½ in. The machinery and boilers down below are thus protected in some measure. Most of these gratings are prevented from lifting as shown, but some have to be hinged for access below, and these are fitted with balance weights to make the lifting easy.

Splinter nettings are provided below the armour bars over the engine-room, so as to protect the machinery from *débris* that might get through the armour gratings. The netting is about 12 in. below the gratings.

Cofferdams.—Around all openings in the middle deck which are necessarily open in action a dwarf bulkhead is carried some feet above the L.W.L. A similar bulkhead is run close to the upper coal-bunker bulkhead (see Figs. 12 and 22). The space thus formed, about 12 in., is termed a *cofferdam*, and into these cofferdams canvas, oakum, or other such material can be jammed down to act as a leak stopper, and so limit the flow of water across the deck, supposing the sides and bulkheads pierced. This is the primary function of these cofferdams, but in many ships they are made rather wider, about 18 in., in order to allow bags and hammocks to be stowed.

Armour Backing and Supports.—In the early days of armour protection with plates of wrought iron, the wood backing was very thick, the object being to provide a support which should be somewhat elastic. This thick backing was continued to the Admiral class, in which the 18-in. armour had 15-in. backing behind. With modern plates, however, it has been found that the best results have been obtained with a perfectly rigid support, and for land fortifications granite has been used. This is manifestly impossible for ships on account of weight, but a massive system of framing is always provided behind armour, with a small thickness of teak backing, about 4 in., to form a bed for the back of the armour. In all cases the skin plating behind armour is in two thicknesses. Figs. 142 and 143 show the framing behind the 18-in. armour of the *Royal Sovereign*. The 4-in. steel above is supported by 6-in. zed bars. In *Majestic* the 9-in. armour has 15-in. plate frames worked every 24 in., with horizontal stiffening girders as Fig. 18. In *Cressy* the 6-in. armour is supported by 10-in. zed bars worked every 24 in., with a horizontal girder as shown in Fig. 144. The battery armour fitted in the latest ships is backed and supported by framing. Barbettes, although

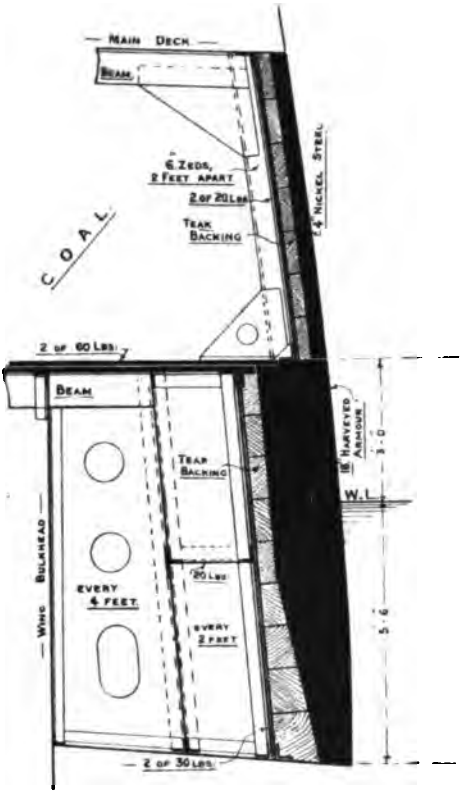


Fig. 142.

Fig. 143.

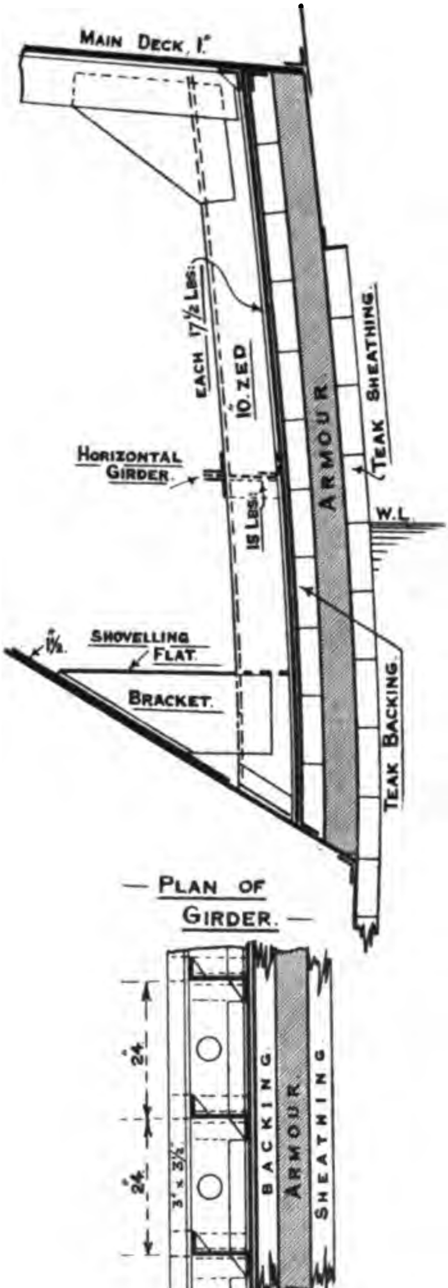


Fig. 144.—Support to 6-in. armour.

well fitted to withstand blows by their shape, are supported inside the double thickness of plating by closely spaced vertical girders.

Behind armour, where men are likely to be employed in action, the inside of the framing is covered in with 10-lb. plating (see

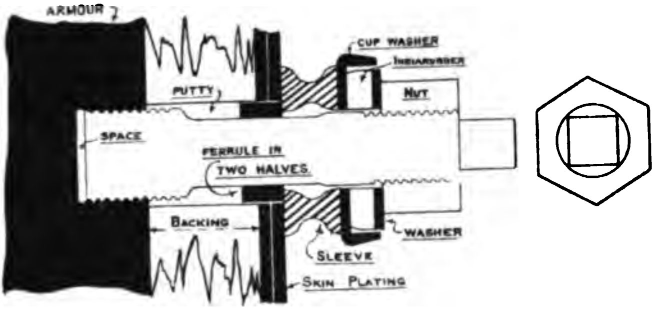


FIG. 145.—Armour bolt.

main deck in Fig. 13). When armour is struck, rivets are likely to break and the heads to fly off, so that this lining forms some protection to men inside. This plating itself should be secured by screw rivets to the framing (see K, Fig. 9).

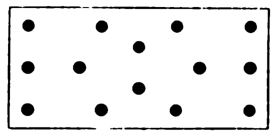


FIG. 146.

Armour Bolts. — With wrought-iron armour, the bolts for securing the armour to the ship's structure were carried right through with a large conical head flush with the surface of the armour. With hard-faced armour, however, the surface must not be pierced for bolts because the surface would then be liable to crack badly from hole to hole when struck. Armour bolts are now screwed into the back of the plate (Fig. 145), and about one bolt to every 7 square ft. is allowed. Fig. 146 shows the holes in a specimen plate; it is important to have good security in order to keep fragments together, even if the plate is badly cracked. Experiments have shown that the pieces are still very efficient, provided they are held up to the backing.

In order to diminish the liability of bolts breaking under the impact of projectiles, the shank of the bolt is made slightly less in diameter than at the bottom of the thread. The bolt will then stretch at this weakest part rather than break under the thread. A sleeve is fitted to provide sufficient length for this weakest

portion; also, to provide some elasticity to take the shock, the nut securing the bolt to the ship is fitted with an elastic washer of india-rubber. This is placed inside a "cup washer" to keep the

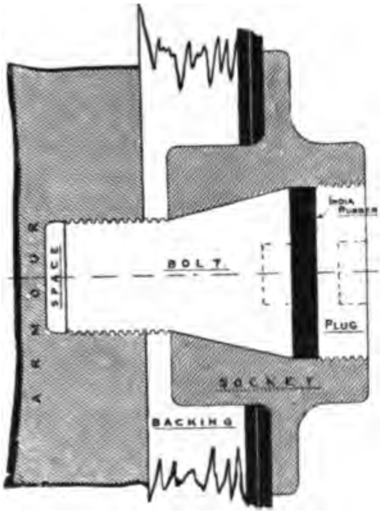


FIG. 147.—Armour bolt.

rubber washer in place. In some places, as barbettes, it is not possible to get sufficient room for an ordinary armour bolt; in such places, the bolt, as in Fig. 147, has to be used.

CHAPTER XIV.

RULES OF MENSURATION FOR THE CALCULATION OF AREAS AND VOLUMES.

1. *Area of rectangle,* as ABCD (Fig. 148).—The area is given by AB $\times$ BC. The length and breadth must be of the same denomination, *i.e.* if the length is in feet the breadth must be in feet, the area then being square feet.

2. *Area of triangle,* as ABC (Fig. 149).—We draw CD perpendicular to the base AB, meeting it, or it produced, in D. Then the area $= \frac{1}{2} \times$ AB $\times$ CD, or one-half the base into the height.

3. *Area of trapezoid.*—This is a four-sided figure, in which

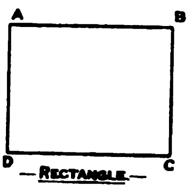


FIG. 148.

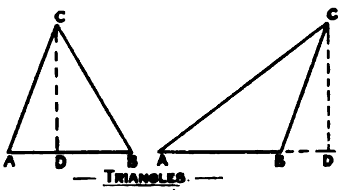


FIG. 149.

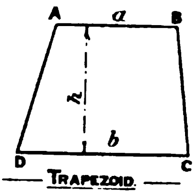


FIG. 150.

two sides only are parallel, as ABCD (Fig. 150). Calling the parallel sides a and b respectively, and h the distance between them, the area $= \frac{1}{2}(a + b)\,h$, or one-half the sum of the parallel sides multiplied by the distance between them.

4. *Circle.*—(*a*) Length of circumference is π times the diameter, where $\pi = 3{\cdot}1416$, or $\frac{22}{7}$ nearly.

(*b*) Area of circle of diameter $d = \dfrac{\pi d^2}{4}$.

5. *Curvilinear figure,* as ABCD (Fig. 151).—The area of figures of this character are continually required in ship calculations.

(*a*) *Trapezoidal rule.*—This rule has found considerable favour, especially in France and the United States, on account of its great

simplicity. We divide the base into a number of equal spaces and erect ordinates to meet the curve as EF, GH, etc. Calling the common interval h and the length of ordinates $y_1, y_2, \ldots y_7$, we have, regarding the figures ADFE, EFHG, etc., as trapezoids—

Area ADFE $= \frac{1}{2}(y_1 + y_2)\,h$
Area EFHG $= \frac{1}{2}(y_2 + y_3)\,h$, and so on to
Area NOCB $= \frac{1}{2}(y_6 + y_7)\,h$.

Adding all together—

$$\text{Area ABCD} = \tfrac{1}{2}h\,(y_1 + 2y_2 + 2y_3 + 2y_4 + 2y_5 + 2y_6 + y_7)$$
$$= h\left(\frac{y_1 + y_7}{2} + y_2 + y_3 + y_4 + y_5 + y_6\right),$$

i.e. the first and last ordinates are added together and divided by

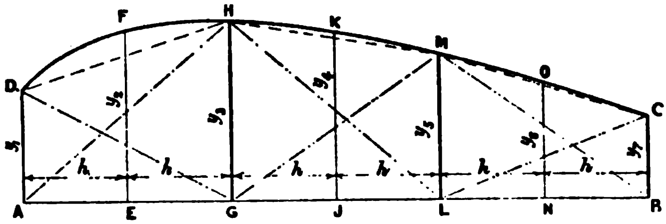


Fig. 151.

two, then all the remaining ordinates are added and the total sum is multiplied by the common interval.

The following example will illustrate the use of the rule and how near the result obtained is to the real area required.

The curve, whose ordinates 2 ft. apart are 0, 2·2, 4·0, 5·4, 6·4, 7·0, and 7·2 ft. respectively, is a portion of a common parabola, and the exact area enclosed is 57·6 square ft. Find the area by using the trapezoidal rule.

$$\text{Area} = 2\left(\frac{0 + 7{\cdot}2}{2} + 2{\cdot}2 + 4{\cdot}0 + 5{\cdot}4 + 6{\cdot}4 + 7{\cdot}0\right)$$
$$= 57{\cdot}2 \text{ square ft.}$$

There is thus an error of nearly 1 per cent. The error involved in using this rule is lessened by spacing the ordinates closely, but it is not used in Admiralty calculations on account of the approximate nature of the results obtained. The rule that is employed is—

(*b*) *Simpson's first rule.*—Take a figure as ABCD (Fig. 152),

and divide the base into two equal parts in the point E. Then, assuming the curve is a common parabola, the area ABCD is given by $\frac{h}{3}(y_1 + 4y_2 + y_3)$. To apply this rule to a longer figure, as ABCD (Fig. 151), we divide the base into an even number of intervals, so that the above rule may be applied to each portion containing a pair of intervals, thus—

$$\text{Area ADHG} = \frac{h}{3}(y_1 + 4y_2 + y_3)$$

$$\text{Area GHML} = \frac{h}{3}(y_3 + 4y_4 + y_5)$$

$$\text{Area LMCB} = \frac{h}{3}(y_5 + 4y_6 + y_7)$$

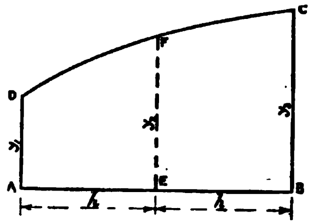


FIG. 152.

Combining together we have—

$$\text{Area ABCD} = \frac{h}{3}(y_1 + 4y_2 + 2y_3 + 4y_4 + 2y_5 + 4y_6 + y_7)$$

The multipliers thus are 1, 4, 2, 4, 2, 4, 1, and there must be an even number of intervals or an odd number of ordinates for the rule to be applicable. In working by this rule it is advisable to use a table as follows, which is the calculation for the area of the figure considered above, the exact area of which is known to be 57·6 square ft.

Number of ordinate.	Length of ordinate.	Simpson's multiplier.	Product.
1	0	1	0
2	2·2	4	8·8
3	4·0	2	8·0
4	5·4	4	21·6
5	6·4	2	12·8
6	7·0	4	28·0
7	7·2	1	7·2
			86·4

$\frac{1}{3}$ common interval $= \frac{2}{3}$
Area $= 86·4 \times \frac{2}{3} = 57·6$ square ft.

The area given by using this rule is seen to be the exact area. This is necessarily so because the curve is a common parabola, and this is the curve on which the rule is based.

EXAMPLE.—The following ordinates, 1·3 ft. apart, give a curve which is an arc of a circle of 6 ft. radius, viz. 0, 1·56, 2·41, 2·86, 3·00, 2·86, 2·41, 1·56, 0. The exact area of the circular segment thus obtained is 22·1 square ft. If the area be calculated by the two methods considered above we have—

(*a*) Area by trapezoidal rule, 21·64 square ft.
(*b*) Area by Simpson's rule, 22·1 „ „

It is thus seen that the latter gives a correct result, even though the curve is not a parabola. Also the trapezoidal rule is in error to the extent of over 2 per cent.

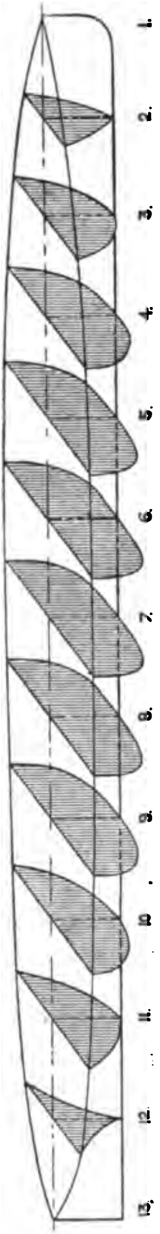


FIG. 154.

Volumes.—To find the volume of a solid bounded by a curved surface. The volumes of such bodies as this are continually required

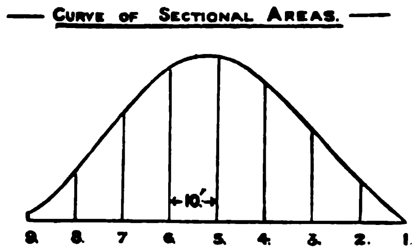


FIG. 153.

in ship calculations, the most important case being the underwater volume of a ship.

The method adopted is to divide the volume by a series of equidistant planes; we then find the area of each of the figures traced out on these planes by the surface, as in Fig. 154, and treat these areas as the ordinates of a curve having the same length as the body. The area enclosed by this "curve of areas" will give the volume required.

In finding the underwater volume of a ship we may divide the volume in two ways, viz.—

1. By means of equidistant planes in transverse sections. The shape of these sections is given in the "body plan" in Fig. 155, and in perspective in Fig. 154.

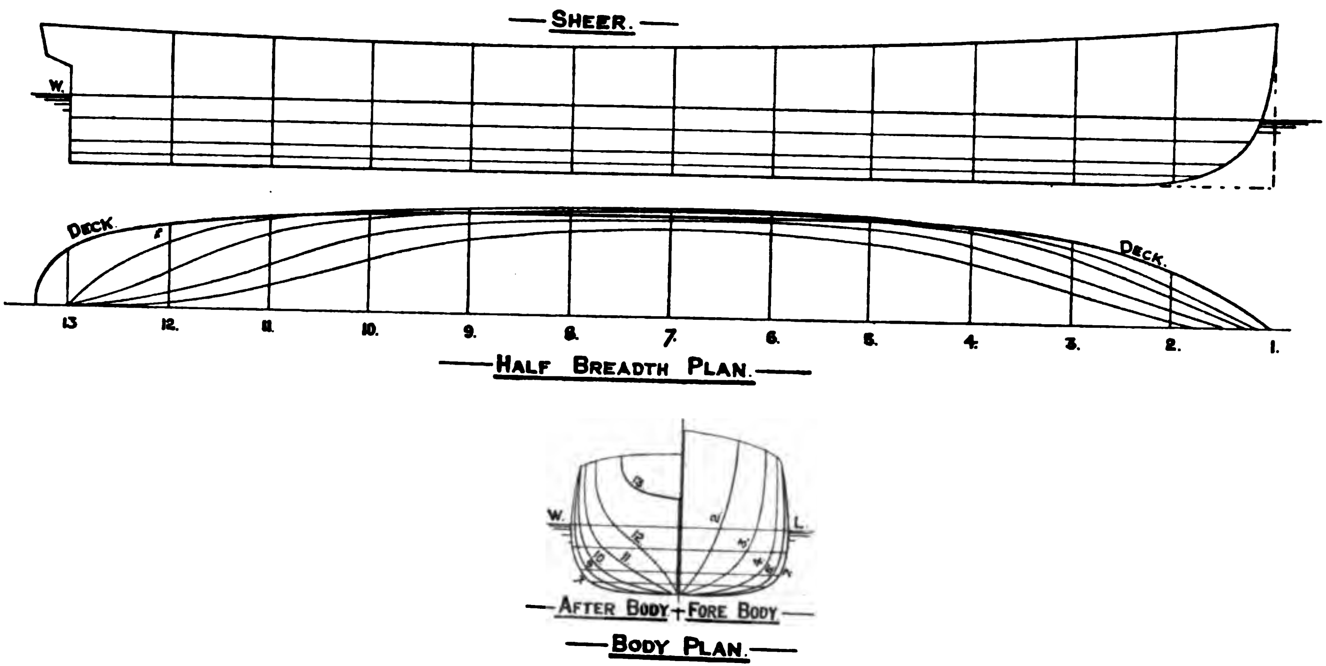

SHEER.
W.
DECK.
DECK.
13
12.
11.
10.
9.
8.
7.
6.
5.
4.
3.
2.
1.
HALF BREADTH PLAN.
W.
L.
13
12
11
10
AFTER BODY
FORE BODY
BODY PLAN.

FIG. 155.

2. By means of equidistant planes in horizontal sections parallel to the load water-plane. (In the ship, Fig. 155, it has been found necessary to introduce a "half" waterline at the bottom.)

In calculating out the underwater volume of a ship both these methods are adopted. An excellent check is thus obtained on the accuracy of the work, because the volume as found by either method should be the same. The following example will illustrate this—

The underwater body of a yacht is divided by transverse planes 10 ft. apart, and the following are the areas, viz.—

0·3, 22·7, 48·8, 73·2, 88·4, 82·8, 58·7, 26·2, 3·9 square ft.

The same body is divided by horizontal planes, 1 ft. 6 in. apart, having the following areas, viz.—

944, 795, 605, 396, 231, 120, 68, 25, 8 square ft.

To find the volume—

The following is the calculation using the areas of the transverse sections, the the curve of areas being given in Fig. 153—

Number of section.	Area.	Simpson's multiplier.	Product.
1	0·3	1	0·3
2	22·7	4	90·8
3	48·8	2	97·6
4	73·2	4	292·8
5	88·4	2	176·8
6	82·8	4	331·2
7	58·7	2	117·4
8	26·2	4	104·8
9	3·9	1	3·9
			1215·6

$\frac{1}{3}$ common interval $= \frac{10}{3}$

Volume $= 1215·6 \times \frac{10}{3} = 4052$ cubic ft.

The following is the calculation using the horizontal sections—

Number of plane.	Area.	Simpson's multiplier.	Product.
1	944	1	944
2	795	4	3180
3	605	2	1210
4	396	4	1584
5	231	2	462
6	120	4	480
7	68	2	136
8	25	4	100
9	8	1	8
			8104

$$\tfrac{1}{3} \text{ common interval} = \frac{1 \cdot 5}{3}$$

$$\text{Volume} = 8104 \times \frac{1 \cdot 5}{3} = 4052 \text{ cubic ft.}$$

A similar calculation is made to find the volume of a coal-bunker.

EXAMPLE.—A coal-bunker has sections $17\frac{1}{2}$ ft. apart, and the areas of these sections are 98, 123, 137, 135, and 122 square ft. respectively. The following calculation will determine the volume—

Area of section.	Simpson's multiplier.	Product.
98	1	98
123	4	492
137	2	274
135	4	540
122	1	122
		1526

$$\tfrac{1}{3} \text{ common interval} = \tfrac{35}{6}$$
$$\text{Volume} = 1526 \times \tfrac{35}{6} = 8902 \text{ cubic ft.}$$

Reckoning 43 cubic ft. of coal to the ton, the capacity of the bunker $= \frac{8902}{43}$ $= 207$ tons.

The capacity of each coal-bunker on board ship is calculated in this way, the height being taken to the under side of beams, and the capacity in cubic feet thus found is marked in some conspicuous place on the bunker bulkhead.

For Welsh coal 40 cubic ft. per ton is taken, for North Country coal 43 cubic ft. per ton.

CHAPTER XV.

NAVY LIST DISPLACEMENT, TONNAGE, ETC.

Length.—The length of ships of the Royal Navy stated in all official documents is always the *length between perpendiculars.* The forward perpendicular is a vertical through the intersection of the fore side of stem with the normal load waterline, the ship being supposed to be floating at this line (see Figs. 67 and 68). The after perpendicular (*a*) in ships with the rudder hinged at the fore side, is taken as the after side of the sternpost, as in Figs. 69 and 75; (*b*) in ships with balanced rudders, it is taken as the centre line of the rudder-head (see Figs. 71, 72, and 79).

The length on the load waterline includes the overhang of the stern at this line. In the United States Navy, for instance, this is the length used in stating a ship's dimensions. Comparisons between ships are apt to be misleading if the lengths are not taken on the same basis. It is believed that the usual French practice is similar to that in the Royal Navy.

The length of ship over all includes the overhang of the stern and the projection of the ram; the length for docking purposes includes also the overhang of the stern walk, if any.

Breadth.—The breadth stated is the breadth of the hull at the broadest part as designed. It sometimes happens that the actual breadth as built is slightly greater than this. For docking purposes the projection of casemates, bilge keels, etc., must be considered, as also the shape of the dock and dock entrance.

Navy List Displacement.—This is always used in official documents, and is a figure which attaches to the ship so long as she remains in the Navy. It is the total designed weight, including the *estimated* weight of hull, machinery, armour, and armament, *legend* weights of water, stores, and coal, and a weight appropriated to a Board Margin. The bunkers in this condition

are assumed about half full of coal; it thus represents a mean condition of the ship.

Draught.—The draught of water corresponding to this Navy List displacement is the *normal load draught.*

It does not follow that, when the ship is finished, she will, with legend coal, etc., exactly float at the designed load water-line. For instance, the Board Margin may not be appropriated, or only a portion of it. The weights of hull, machinery, armour, or armament may turn out greater or less than estimated at the time of the design. Ships of the same class, built at different yards, from the same drawings and specification, sometimes differ among themselves by considerable amounts.

The draught marks are not usually set up on the ship at the perpendiculars, but at the points where the keel cuts up at the bow and stern. In the special case of destroyers and vessels with propellers below the line of keel, a set of draught marks are set up on the shaft brackets, showing the draught at the bottom of propeller sweep. The draught marks at the after cut up of the keel are used for docking purposes only.

The draughts of water of each ship in the Navy, as completed, in three conditions of the ship, are given in the stability statement furnished to the ship's book.

(*a*) *The normal load draught,* the ship being fully equipped, with reserve feed-tanks empty, and with the legend weights of coal, etc., on board.

(*b*) *The deep load draught,* the ship being fully equipped with fresh-water and reserve feed-tanks full and bunkers full.

(*c*) *The light draught* being an extreme light condition of the ship. All coals, water (including reserve feed), provisions, officers' stores and slops, and one-half the carpenter's, boatswain's, and engineer's stores are assumed to be consumed. No expenditure of ammunition or shell is assumed for this condition.

For vessels with considerable sail, like the sloops, which are likely to proceed under sail alone, the light condition is taken as above, but with the boilers quite full, and the engine condensers and feed-tanks empty.

A specimen stability statement is given at the end of Chapter XIX.

It should be stated that ships generally increase in draught somewhat as time goes on, owing to the alterations and additions carried out. Large weights of paint are often worked into a ship, one coat succeeding another, until the weight of the whole is

very considerable.[1] The question of the draught is looked into occasionally as required, and when necessary a new stability statement is issued. This is specially done when a ship has undergone an extensive refit.

The position of the *deep load line* is indicated inside the ship by label plates and a broken paint line for the information of the ship's officers.

Trials.—The steam trials of a ship, carried out to ascertain how far the speed estimated at the time of the design has been realized, are run at the normal load draught corresponding to the Navy List displacement. Ships are sometimes tried for this purpose when in an incomplete condition, and they are then ballasted to give the required draught of water. If the trial is, however, simply to determine the power developed, and for the acceptance of the machinery, it is not necessary to bring the ship to the load line, provided that sufficient immersion is obtained for the propellers. Except in special ships, as in destroyers, etc., contractors are not responsible for speed, but only for the power developed under the given conditions. It has been found a great convenience, in some cases, to finish the contractor's trials as early as possible, so that the opening up of the machinery, to determine the final acceptance, may proceed while the ship is being completed in other respects.

Tonnage.—We have seen that war-ships are known by their displacement tonnage, this being the total weight as designed with legend weights on board. This tonnage is specially suitable for war-ships, as these ships have to carry a fixed load of armour, guns, etc. For merchant ships, however, a different system is adopted; here the tonnage is a measure of the internal capacity of the ship.

Gross tonnage is the total closed in capacity of the ship, excluding double bottoms, reckoned in tons of 100 cubic ft. The *nett or register tonnage* makes certain deductions from the gross for the space occupied by the crew, etc., machinery and coal. The intention is that the nett or register tonnage shall give a measure of the earning capacity of the ship for carrying cargo and passengers.

It is necessary that ships of the Royal Navy should be measured for their tonnage, in order to form a basis for the payment of dues and other charges at foreign ports. A tonnage

[1] In one case the paint removed from the crew space of a destroyer weighed over 2 lbs. per square foot.

certificate is issued to all the ships of the Navy from measurements made by the officers of the Board of Trade. There are two systems, viz. the British system and the Danube system. They differ in the allowances made for deck erections and machinery spaces. The following shows how these compare with one another, and how the gross tonnage compares with the displacement tonnage in several cases. The Danube rule is the one used when passing through the Suez Canal.

	Displacement tonnage.	Register tonnage.		Gross tonnage.	
		British.	Danube.	British.	Danube.
Battle-ship	14,900	5,796	5,269	8,523	8,609
First class cruiser	11,000	3,610	3,307	7,069	7,123
Second class cruiser	5,600	1,919	1,770	3,770	3,812
Third class cruiser	2,135	713	684	1,492	1,520

It is seen that the gross tonnage in all cases is considerably less than the displacement tonnage.[1]

The following extract from the King's regulations gives the instructions concerning the question of tonnage.

The register tonnage according to British rule is to be inserted in all pilotage certificates, and is to be the basis of all tonnage payments at foreign ports by H.M. ships, except when entering Port Said or the Suez Canal, in which case the tonnage according to the Danube rule is to be issued.

The Board of Trade tonnage certificate, which shows the registered tonnage according to both rules, is furnished to all ships as they are commissioned at home ports.

The weight in tons shown in the Navy List is in no case to be used for the payment of pilotage, nor to be mentioned in pilotage certificates.

[1] In the Atlantic liner *Campania* the displacement tonnage is 18,000 tons, the gross tonnage 12,950, and the register tonnage 4973. The new Cunarders are stated to be designed for a displacement of 32,000 to 33,000 tons, or twice that of *King Edward VII.* The different systems of reckoning tonnage (viz. war-ships by their displacement and merchant ships by their capacity in tons of 100 cubic feet), has obscured the great increase in size of the large liners as compared with the largest war-ships.

CHAPTER XVI.

BUOYANCY, DISPLACEMENT, TONS PER INCH, ETC.

Buoyancy.—At every point of the surface of a body immersed in water there is a pressure which acts normally to the surface. The amount of this pressure will depend on the depth of the point below the surface. If d be the depth of the point below the surface in feet, and w the weight of a cubic foot of water, then the pressure per square foot is $w \times d$ lbs. Thus if a hole of 1 square ft. is made in a ship's bottom $17\frac{1}{2}$ ft. below the surface of the water, a piece of wood would have to be held against the hole with a force of $\frac{64 \times 17 \cdot 5}{2240} = 0 \cdot 5$ ton to keep the water out. It is because of this pressure that diving operations beyond a certain depth are rendered impossible.

In the case of a floating body like a ship, these normal pressures all over the surface act in many different directions. In each case, however, the normal pressure may be resolved into its three components at right angles, viz. (i.) horizontal in a fore-and-aft direction, (ii.) horizontal in a transverse direction, and (iii.) vertical. If the ship is floating at rest, all these horizontal components must balance between themselves, since there is no bodily movement of the ship in any direction. It is the combined effect of all the vertical components which exactly balances the weight of the ship. The single vertical force, which is the resultant of an infinite number of small vertical forces acting on the ship, is termed the *buoyancy*. In the same way the forces due to the weights composing the ship have a single resultant, which we term the *weight* of the ship. The vertical forces acting on the ship at rest are therefore—

(*a*) The weight of the body acting vertically down.
(*b*) The buoyancy acting vertically up.

Since the body has no motion up or down, it follows that *the buoyancy exactly equals the weight.*

If we take a block of wood, say pitch-pine or teak, and suspend by a spring balance, the dial would show the weight of the wood, say 10 lbs. If we place the block into water we should notice, as the wood descends, that the dial registers less and less, showing that the water is taking some of the weight. A point will at last be reached when the dial will register zero, and then the wood is floating, and instead of the 10 lbs. being borne partly by the balance and partly by the water, the whole weight is taken by the water. That is, the buoyancy is equal to the weight.

Displacement.—We now come to a most important proposition in connection with floating bodies. **A body floating freely and at rest displaces or puts aside a volume of water having a weight exactly equal to the weight of the body.** The water displaced is termed the *displacement,* and can be either reckoned as a volume, when it is expressed in cubic feet, or as a weight,

FIG. 156.

when it is expressed in tons. That the above proposition must be true may be seen by the following:—

Consider a vessel floating freely and at rest in still water, and imagine if it were possible that the water is solidified, maintaining the same level, and therefore the same density. If now we lift the vessel out we shall have a cavity left behind, which is exactly of the form of the underwater volume of the ship, as Fig. 156. Now suppose the cavity is filled with water. This amount of water is evidently the displacement of the vessel. Suppose now that the solidified water outside again becomes water. The water we have poured in will be supported by the water surrounding it. The support given, first to the vessel and now to the water we have poured in, by the surrounding water must be the same, and consequently it follows that the weight of the vessel exactly equals the weight of the water poured in to fill the cavity, or, in other words, *the weight of the vessel is equal to the weight of the water displaced.*

This fact is of immense assistance in dealing with the weights of ships. We do not need to estimate the weights, which would be an almost impossible task, but, knowing the line at which the ship is floating, and having the drawings giving the shape of the ship, we can calculate the volume of displacement up to that line. Then, knowing the density of the water, we at once have the weight of the ship with everything she has on board.

When a man is floating in water, it is manifestly desirable to keep the arms below the surface because of the buoyancy due to their displacement. It will be noticed that, when thus floating, if the arms are held up out of the water, a certain amount of sinkage occurs. The weight of the body remains the same, but the buoyancy is reduced and must be made up by the sinkage before the balance is obtained, viz. that the weight equals the buoyancy.

The same principle has to be borne in mind when constructing a raft. All the human beings have to be placed *on* it, but a great quantity of provisions, etc., may be safely carried *under* it. For instance, a cask of beef weighs 300 lbs., and its volume displaces 184 lbs. of water, so that if carried beneath the raft we get 184 lbs. of buoyancy from it, the net weight of the cask is therefore only 116 lbs.

The density of water varies at different places, and often at the same place at different states of the tide. Thus in places on the coast and at sea it is practically constant, viz. 64 lbs. per cubic foot, giving 35 cubic ft. to the ton. At Glasgow, well up the Clyde, the water weighs 62½ lbs. per cubic foot, giving 35·84 cubic ft. to the ton. At Gravesend the water is 63·7 lbs. per cubic foot at high tide, and 63·4 lbs. per cubic foot at low tide. It is because of the difference of density that a vessel *decreases* her draught in going from fresh to salt water. This is of importance in merchant vessels, which are allowed to be loaded below the load line disc, when floating in water less dense than salt. Thus a vessel of 20 ft. depth, if in fresh water (1000 ozs. to the cubic foot), is allowed by the Board of Trade officers to be loaded 4 in. deeper than the load line disc, because it is known that when she has got to sea she will rise this amount. If the same ship were being loaded in Aberdeen Harbour, for instance, where the water weighs 1015 ozs. to the cubic foot, 1½ in. only would be allowed.

The vessel whose displacement has been calculated in the previous chapter as 4052 cubic ft., will weigh if floating at the top waterplane in salt water $\frac{4052}{35} = 115{\cdot}8$ tons. If floating at the same waterplane in river water, of which 35·6 cubic ft. go to the ton, the weight would be $\frac{4052}{35{\cdot}6} = 114$ tons.

Curve of Displacement.—We have seen how to determine the volume of displacement of a ship when floating at a given waterplane. The draught of a ship, however, continually varies owing to having different weights of coal, stores, etc., on board, and so it is desirable to have the means of determining quickly the displacement of a ship at any other given draught. The body having been cut by a series of equidistant planes, we calculate the displacement to each of these planes in succession. Thus in a battle-ship the following figures were obtained at planes 4 ft. 3 in. apart, the top one being at a draught of 31 ft., viz. 18,300, 15,016, 12,121, 9,293, 6,510, 3,923, 1,901, 268 tons respectively. We can then set up a scale of draughts, as in Fig. 157, and at each line set out the corresponding displacement. Through the spots thus given we then draw a curve, which is termed the *curve of displacement.* In this case, suppose the ship is floating at a draught of 19 ft. 6 in. forward and 20 ft. 10 in. aft, *i.e.* 20 ft. 2 in. mean. We set up this draught at AB and measure to the curve, and find the displacement to be 10,550 tons. It is usual to continue the curve right down to zero draught, although the ship could never float at a less draught than would be due to her structure alone. The smallest displacement obtained in the history of any ship would be when she was launched.

Tons-per-inch Immersion.—It is frequently necessary to know how much a vessel, when floating at a given draught (*a*) will sink, if certain known weights are put on board, or (*b*) will rise if certain known weights are removed. Since the displacement of a vessel equals the weight, any extra displacement caused by adding a weight must equal the added weight. If A is the area in square feet of the waterplane at which a ship is floating, then the volume of a layer 1 in. thick is $\frac{A}{12}$ cubic ft., and the displacement of this layer is $\left(\frac{A}{12}\right) \div 35 = \frac{A}{420}$ tons. This must be the weight necessary to add to sink the ship 1 in., or to take out to lighten the ship 1 in. This is termed the *tons-per-inch immersion.*

EXAMPLE.—The area of the waterplane at which a vessel is floating is 7854 square ft. Find the rise due to burning 56 tons of coal.

$$\text{Tons per inch} = \frac{7854}{420} = 18{\cdot}7 \text{ tons}$$

$$\text{Rise} = \frac{56}{18{\cdot}7} = 3 \text{ in.}$$

Approximation to the Value of the Tons-per-inch Immersion.—If L is the length between perpendiculars, and B is the breadth of the hull—

Then (1) for ships with fine L.W.P. as cruisers—

$$\text{Tons per inch} = \frac{(L \times B)}{600} \text{ approximately.}$$

(2) for ships of fuller form as battle-ships—

$$\text{Tons per inch} = \frac{(L \times B)}{530} \text{ approximately.}$$

These approximations will be found useful in the absence of the correct figure for a particular ship. The correct tons per inch is usually given in the Ship's Book. The length between perpendiculars and breadth of hull are used above instead of the length and breadth on L.W.L., as the latter are not usually known without reference to the drawings.

The following show how far the approximation holds in different ships:—

Third class cruiser	actual	18·7,	approximation	18·25
First class cruiser	,,	58	,,	59·2
Battle-ship of fine form	,,	54·2	,,	57·7
Battle-ship of full form	,,	56·7	,,	55·2

Curve of Tons-per-inch Immersion.—The above figures refer to the tons per inch at the L.W.L. In order to determine the tons per inch at any other line at which a ship happens to float we construct a tons-per-inch curve. We calculate the tons per inch at the several level lines parallel to the L.W.L., and set these out at the corresponding draughts on a convenient scale. Thus, for a battle-ship, the tons per inch at level lines 4 ft. 3 in. apart, commencing at 31 ft. draught, were found to be respectively—57·5, 57·2, 56·5, 53·7, 51·0, 46·3, 39·0, 24·0. The curve drawn through the spots thus obtained, as in Fig. 157, is the *curve of tons-per-inch immersion.* If the vessel floats at a draught of 19 ft. 6 in. forward and 20 ft. 10 in. aft, *i.e.* 20 ft. 2 in. mean, this draught is set up and the ordinate of the curve, 54·8, is the tons per inch required.

In ordinary ships the value of the tons per inch varies very little for considerable changes of draught in the neighbourhood of the L.W.L.

Coefficient of Fineness.—This is the ratio which the actual volume of displacement bears to the volume of a rectangular block having the same length as that of the ship between perpendiculars,

the same breadth and the depth equal to the mean draught of the ship (the draught should be excluding any keel projection, if any, as in a sheathed ship).

The value of this coefficient gives us a good idea of the degree of fineness of a ship, as the following examples show:—

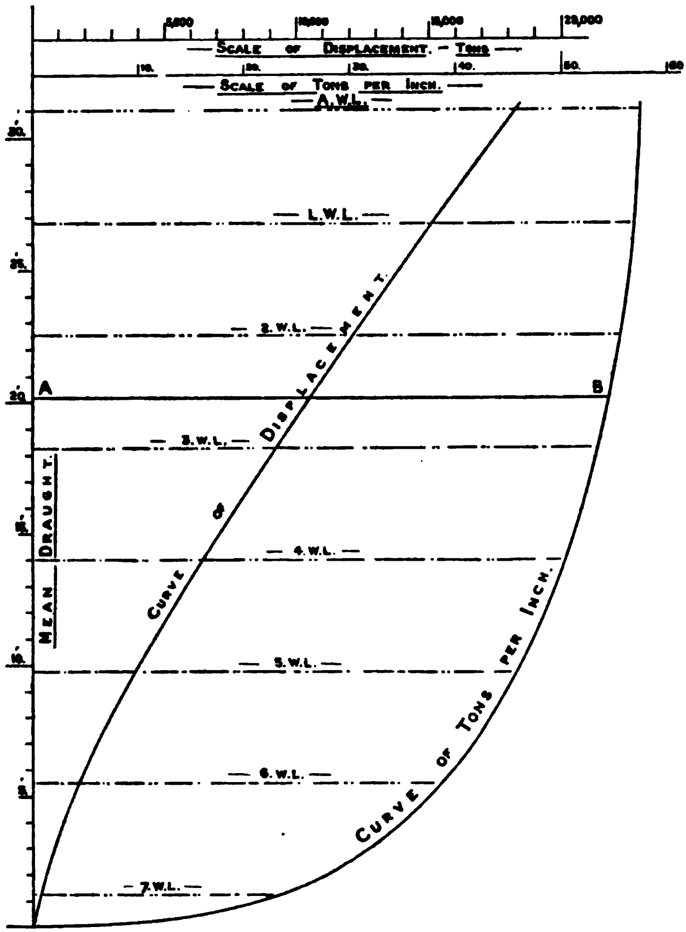


FIG. 157.

"**Formidable.**"—400 ft. × 75 ft. × $26\frac{3}{4}$ ft. × 15,000 tons; 15,000 I.H.P., 18 knots.

$$\text{Volume of displacement} = 15{,}000 \times 35 \text{ cubic ft.}$$
$$\text{Volume of block} = 400 \times 75 \times 26\cdot 75 \text{ cubic ft.}$$
$$\textit{Coefficient of fineness} = \frac{15{,}000 \times 35}{400 \times 75 \times 26\cdot 75} = 0\cdot 65.$$

"**Duncan.**"—405 ft. × 75½ ft. × 26½ ft. × 14,000 tons; 18,000 I.H.P., 19 knots.

$$\textit{Coefficient of fineness} = \frac{14{,}000 \times 35}{405 \times 75{\cdot}5 \times 26{\cdot}5} = 0{\cdot}6.$$

"**Amethyst.**"—360 ft. × 40 ft. × 14½ ft. × 3000 tons; 9800 I.H.P., 21¾ knots.

$$\textit{Coefficient of fineness} = \frac{3000 \times 35}{360 \times 40 \times 14{\cdot}5} = 0{\cdot}5.$$

"**Drake.**"—500 ft. × 71 ft. × 26 ft. × 14,100 tons; 30,000 I.H.P., 23 knots.

$$\textit{Coefficient of fineness} = \frac{14{,}100 \times 35}{500 \times 71 \times 26} = 0{\cdot}53.$$

The following are average values of this coefficient of fineness, viz.—

Battle-ships	0·6	to	0·65
Cruisers	0·5	„	0·55
Destroyers	0·4	„	0·45

As illustrating the importance of the fineness of a ship in connection with the attainment of speed, reference may be made to the *Formidable* and *Duncan.* In the former ship a coefficient of fineness of 0·65 was adopted, 15,000 I.H.P. being intended to realize 18 knots. In the latter ship, in order to attain 19 knots without an excessive expenditure of I.H.P., a finer form had to be adopted with a coefficient of fineness of 0·6.

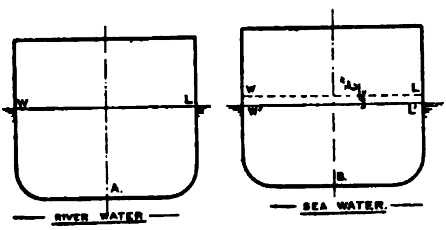


Fig. 158.

Difference of Draught of Water when Floating in Salt Water and River Water.—(Salt water 64 lbs., river water 63 lbs. to the cubic foot.) The weight of the ship going from the one to the other remains the same = W tons = W × 2240 lbs.

$$\text{Volume of displacement in salt water} = \frac{W \times 2240}{64} \text{ cubic ft.}$$

$$\text{„ \quad „ \quad „ river „} = \frac{W \times 2240}{63} \text{ „ \quad „ .}$$

$$\therefore \left.\begin{array}{l}\text{volume of the layer between the}\\ \quad\text{two lines (Fig. 158)}\end{array}\right\} = \frac{W \times 2240}{63} - \frac{W \times 2240}{64}$$

$$= \frac{W \times 2240}{63 \times 64} \text{ cubic ft.}$$

If T be the tons per inch in salt water and t the difference of draught in inches, the weight of the layer $= T \times t \times 2240$ lbs., and the volume of the layer $= \dfrac{T \times t \times 2240}{64}$ cubic ft.

We have thus found the volume of the layer in two ways, and we can then equate, viz.—

$$\frac{W \times 2240}{63 \times 64} = \frac{T \times t \times 2240}{64}$$

$$\text{or } t = \frac{W}{63 \times T}$$

Thus for a battle-ship 57·2 tons per inch, 15,000 tons displacement, the difference of draught $= \dfrac{15{,}000}{63 \times 57{\cdot}2} = 4$ in.

If for W and T we put approximate values, we have—

$$(a)\ \text{Battle-ships}\ t = \frac{(0{\cdot}62 \times L \times B \times D)\,\frac{1}{35}}{63 \times (\frac{1}{530} \times L \times B)}$$

$$= \frac{1}{6{\cdot}7} \times \text{draught.}$$

$$(b)\ \text{Cruisers} \quad . \quad t = \frac{(0{\cdot}52 \times L \times B \times D)\,\frac{1}{35}}{63 \times (\frac{1}{600} \times L \times B)}$$

$$= \frac{1}{7{\cdot}1} \times \text{draught.}$$

We can therefore say, roughly speaking, that the difference of draught in *inches* is one-seventh the draught *in feet*.

In the general case of a ship passing from water of density d' to water of density d (d' being greater than d), the difference of draught is $\dfrac{W}{T} \cdot \dfrac{d'-d}{d}$.

Reserve of Buoyancy and Freeboard.—We have seen that *buoyancy* is the upward support given by the water to the ship, and this upward force exactly equals the weight of the ship.

Freeboard is the height of the upper deck at side (to top of deck plank if fitted) from the water surface. *Reserve of buoyancy* is the volume of the ship above the waterplane which can be made watertight. In many ships this will be to the upper deck, but in some ships there are erections, as poop and forecastle, which can be made watertight. The sum of the buoyancy and the

N

reserve of buoyancy is the total floating power of the vessel. Reserve of buoyancy is expressed as a percentage of the buoyancy, and this varies considerably in different types of ship, *e.g.* in the *Devastation*, a low freeboard ship, this percentage was about 50 per cent. For modern battle-ships with good freeboard it amounts to about 90 per cent., and for cruisers and destroyers higher values than this are usual.

In merchant vessels sufficient reserve of buoyancy is obtained by specifying the minimum freeboard, this being obtained from tables drawn up by the Board of Trade. All British war-ships have freeboard and reserve of buoyancy considerably in excess of what the Board of Trade would require for merchant vessels of corresponding dimensions. Ships like the torpedo gunboats appear to disadvantage in comparison with merchant vessels of similar size, because of the absence of bulwarks which extend several feet above the upper deck of the latter ships. The presence of the bulwarks, although affording protection from the sea, does not, of course, increase the reserve of buoyancy. Indeed, bulwarks are likely to become a source of danger, if provision is not made for a sufficient number of large clearing ports for the purpose of speedily clearing the deck of water.

The reserve of buoyancy possessed by a ship is important, because this has to be drawn upon if a ship is damaged, and a ship with small reserve could only stand a small amount of damage before being entirely submerged. Freeboard and reserve of buoyancy are also important, however, because of the necessity of providing sufficient stability at large angles of inclination. This will be dealt with in a later chapter.

Sinkage caused by a Central Compartment being open to the Sea.—The principles involved will be well illustrated if we take a box-shaped vessel. Such a vessel 100 ft. long, 20 ft. broad, 20 ft. deep floats at a draught of 10 ft. What will be the draught if a central compartment, 20 ft. long, is laid open to the sea (Fig. 159).

The weight of the vessel is the same before and after the bilging, but the buoyancy has been diminished, and the vessel is in a similar condition to one having the watertight side and bottom between the bulkheads replaced by a lattice-work. In consequence of the loss of buoyancy the vessel must draw on the reserve of buoyancy by sinking down to the waterline W′L′ to the draught d feet, say—

Original volume of displacement $= 100 \times 20 \times 10$,
New „ „ „ $= 80 \times 20 \times d$;

and seeing that the weight of ship is the same, these two volumes must be the same, so that $d = 12{\cdot}5$ ft.

It is important to note that if a compartment is filled with coals, stores, etc., the space thus occupied cannot be taken up by the water, and thus the lost buoyancy is much less. For instance, in the above vessel, if the central compartment were filled with coal from the ends, the vessel floating at the same draught of 10 ft. before bilging, the sinking after bilging would only be $9\frac{1}{2}$ in.

If the watertightness of either of the bulkheads in the former example ceased below a height of 12 ft. 6 in., the water would

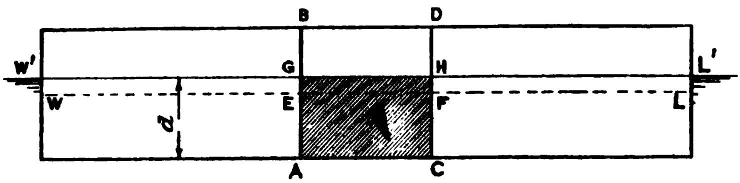


FIG. 159.

not be confined to the central compartment, and the vessel would sink. This illustrates the importance of carrying watertight bulkheads well above the waterline. This is always done with the principal bulkheads in vessels of the Royal Navy (see Figs. 52 and 54).

Watertight flats are important because they serve to confine the results of any damage which may occur. Thus, in the first of the above examples, if a watertight flat were worked 5 ft. from the keel between the bulkheads, if bilging took place

(i.) Below the flat, a sinkage of 1 ft. would result;

(ii.) Above the flat, a sinkage of $1\frac{1}{4}$ ft. would result.

The greater sinkage in the latter case is due to the fact that the waterplane area is reduced as well as the buoyancy.

CHAPTER XVII.

INITIAL STABILITY, METACENTRIC HEIGHT, ETC.

Centre of Gravity.—The weight of a body is the sum or resultant of the weights of all the particles composing it, and this resultant acts through a definite point however the body is placed. This point is termed the *centre of gravity* (C.G.), and the body being at rest, we can regard the whole weight as being concentrated at the centre of gravity.

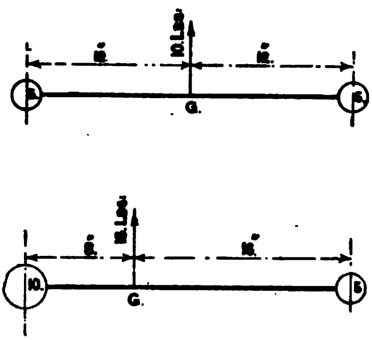


FIG. 160.

If two weights each of 5 lbs. are placed, as in Fig. 160, 24 in. apart, the C.G. of the system must be midway between them at G, and if the weights were held by a string, it would be immaterial, so far as the string was concerned, whether the weights are as shown, or the whole 10 lbs. concentrated at G. Again, if the weights are 10 and 5 lbs. respectively, 24 in. apart, we should need to support at the point G, 8 in. from the larger weight, *i.e.* at the centre of gravity.

The centre of gravity of a ship is the point at which we may regard the whole of the weight to be concentrated.

Centre of Buoyancy.—The resultant of the upward buoyancy must have its line of action through the centre of gravity of the displacement. When the water filled the space, before the ship was there, the *weight* of the water acted through this point, and so the support of the surrounding water, or, as we term it, the *buoyancy*, must also act through the same point. This point is termed the *centre of buoyancy*, being the C.G. of the displacement.

The position of the centre of buoyancy (C.B.), relative to the

waterplane, can be accurately calculated by simple rules[1] (which it is not proposed to discuss in this work), and this position has an important influence on the transverse stability of the ship.

The following formula gives the approximate distance of the C.B. below the L.W.L., viz.—

$\frac{1}{3}\left(\frac{D}{2}+\frac{W}{12\,T}\right)$ where D = mean draught (excluding keel projection, if any)
W = displacement in tons
T = tons per inch

This formula is found to give results very close to the figures obtained from detailed calculation, as the following examples show:—

1. Vessel 2135 tons, 18·7 tons per inch, 13 ft. 6 in. mean draught.

$$\text{C.B. below L.W.L. approx.} = \tfrac{1}{3}\left(\frac{13{\cdot}5}{2}+\frac{2135}{18{\cdot}7\times 12}\right) = 5{\cdot}42 \text{ ft.}$$

The actual calculation gave 5·37 ft.

2. Vessel 15,000 tons, 57·2 tons per inch, 26 ft. 9 in. mean draught.

$$\text{C.B. below L.W.L. approx.} = \tfrac{1}{3}\left(\frac{26{\cdot}75}{2}+\frac{15{,}000}{12\times 57{\cdot}2}\right) = 11{\cdot}76 \text{ ft.}$$

The actual calculation gave 11·77 ft.

Conditions of Equilibrium of a Body floating freely and at Rest in Still Water.—We have seen that for any body floating freely and at rest, the upward support of the buoyancy exactly equals the weight of the body. The weight acts through the centre of gravity, and the buoyancy acts through the centre of buoyancy. If the vessel is in equilibrium, *i.e.* has no tendency of herself to move, these two forces which are acting on the ship must act in the same vertical line. That is to say, the C.G. and the C.B. must be in the same vertical line.

This is the condition of things in a tug of war. If no movement takes place, it is evident that both sides are equal, and they are both pulling *in the same line.*

We have thus two conditions which are satisfied in the case of a body floating in equilibrium, viz.—

1. *The weight of the body exactly equals the weight of the water displaced, and*

2. *The C.G. and the C.B. must be in the same vertical line.*

[1] See the author's "Theoretical Naval Architecture."

Stable, Unstable, and Neutral Equilibrium. — The equilibrium, however, may be either (*a*) stable, (*b*) unstable, or (*c*) neutral. These kinds of equilibrium are defined as follows, viz.—

(*a*) *Stable Equilibrium.*—If the vessel be slightly inclined from her position of rest she will tend to return.

(*b*) *Unstable Equilibrium.*—If the vessel be slightly inclined from her position of rest, she tends to incline still further.

(*c*) *Neutral Equilibrium.*—If the vessel be slightly inclined from her position of rest, she neither tends to return to or to incline still farther from that position.

These three kinds of equilibrium are seen in Fig. 161, in which cylinders are shown resting on a smooth table.

(*a*) In the first case the C.G. is below the centre, and if the cylinder is

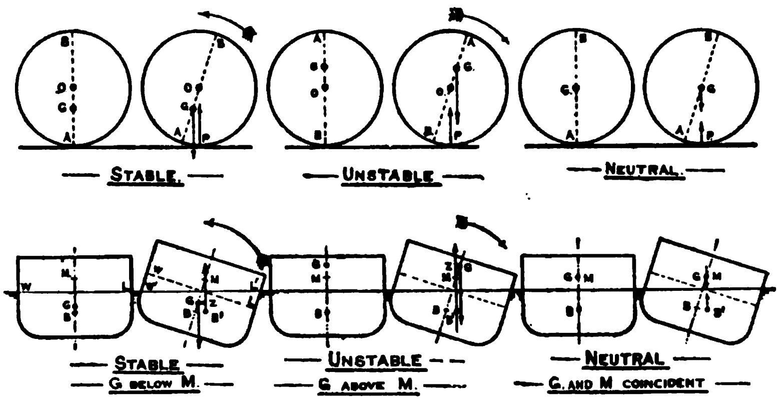


FIG. 161.

slightly inclined it is seen that the two forces acting, viz. (1) the weight through the C.G., and (2) the support of the table through the point of contact, form a couple which tend to take the cylinder back to the upright. This is a state of equilibrium which is *stable.*

(*b*) In the second case the C.G. is above the centre, and on slightly inclining it is seen that the couple acting is an upsetting couple, and the cylinder will incline still farther. This is a state of equilibrium which is *unstable.*

(*c*) In the last case the C.G. coincides with the centre of the figure, and there will be no tendency to return to or incline further from the original position on giving a small inclination. This is a state of equilibrium which is *neutral.* This is the necessary condition for a billiard ball. The ivory must be perfectly homogeneous, so that the C.G. is at the centre of the ball, and the ball must be perfectly spherical, so that the support will always act through the centre.

In the case of a ship let Fig. 162 represent the vessel inclined to a small angle θ. WL was the position of the waterline on the ship when upright; B, the position of the C.B. when upright; and G, the position of the C.G. The ship as inclined has a new waterline, W′L′, and the C.B. of the new displacement will be at B′, so that the buoyancy will now act through B′. Let the vertical through B′ cut the original vertical through B in M.

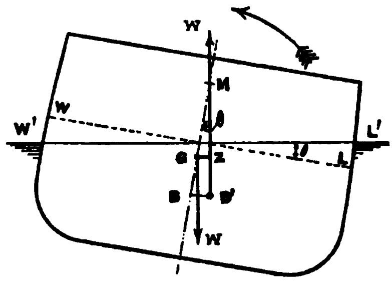


FIG. 162.

Then in the first case, in Fig. 161, we notice that the couple acting on the ship tends to bring her back to the upright. In the second case the couple tends to incline her still further from the upright. We see that whether the couple is a righting couple or an upsetting couple depends on the relative positions of the points G and M.

If G is below M, the couple is a righting couple, and the ship is in *stable* equilibrium. If G is above M, the couple is an upsetting couple, and the ship is in *unstable* equilibrium. If G coincides with M, the ship is in *neutral* equilibrium. The point M is thus seen to be an important point, as its position relative to G determines the state of the equilibrium in the upright condition. It is given the name of the *transverse metacentre* when dealing with transverse inclinations, and the distance between G and M is termed the *metacentric height.*

We thus see that there are three conditions which must be fulfilled in order that a floating body shall float freely and at rest in stable equilibrium, viz.—

1. *The weight of the body must exactly equal the weight of the water displaced;*
2. *The C.G. and the C.B. must be in the same vertical line;* and
3. *The C.G. must be below the transverse metacentre.*

For angles up to 10 or 15° the intersection of the vertical through B′ with the original vertical through B is practically at the same point, viz. M. For larger angles this will not be the case.

Before dealing generally with the question of initial stability, *i.e.* stability at small angles, we shall deal separately with the two

points which determine it, viz. the transverse metacentre and the centre of gravity.

Position of the Transverse Metacentre of a Ship when floating at any given Waterline.—The point M depends solely upon the geometrical shape of the underwater body, and its position can be determined for the ship when floating at any particular waterline.[1] It is determined with reference to the C.B., and because of this the position of the C.B. has a distinct influence on the stability. The distance BM is given by (Moment of Inertia of waterplane about the centre line) ÷ (Volume of displacement) or $BM = \frac{I}{V}$.

The *moment* of a figure about any axis is obtained by dividing the area into a large number of small areas, and multiplying each by its distance from the axis. The addition of all such products is called the *moment.*

The *moment of inertia* of a figure about any axis goes one step further. The area is divided into a large number of small areas, and each of these is multiplied by the *square* of its distance from the axis. The addition of all such products is a moment of the second degree, and is called the *moment of inertia.*[2]

The expression $\frac{I}{V}$ may be approximately found as follows:—

(1) I can be written $I = n.L.B^3$

where L = length of ship

B = breadth of ship

n is a coefficient varying with different shapes of waterplane (for a rectangle $n = \frac{1}{12}$)[2]

[1] See the author's "Text-book of Theoretical Naval Architecture."

[2] This definition and the value of the moment of inertia given for a rectangle may be illustrated by the following approximation to the I of a rectangle 100 ft. × 20 ft. about the centre line. Divide the breadth into ten strips, each having an area of 100 × 2 = 200 square ft.; the centre of the strips from the centre line are 9, 7, 5, 3, and 1 ft. respectively. The I of the half of rectangle about the centre line is therefore nearly

$$[200 \times (9)^2] + [200 \times (7)^2] + [200 \times (5)^2] + [200 \times (3)^2] + [200 \times (1)^2]$$

$$= 200 \times 165 = 33{,}000$$

and for both sides I = 66,000. The exact value is $\frac{1}{12}(100)(20)^3 = 66{,}666$. If the strips had been taken 1 ft. wide a closer approximation still would have been obtained.

(2) V can be written $V = k.L.B.D$

where D = mean draught (ex keel projection if any)

k is a coefficient of fineness

$$\text{so that } BM = \frac{I}{V} = \frac{n.L.B^3}{k.L.B.D} = a.\frac{B^2}{D}$$

where a is a new coefficient obtained from n and k. It is found that a varies between narrow limits, being about $\frac{1}{12}$ in battle-ship forms and $\frac{1}{13}$ in cruiser forms. The important point that this approximate formula emphasizes is, the great influence of beam on the distance BM, and therefore on the position of M. This is the reason why battle-ships have to be so broad. It is necessary to have M above G for stable equilibrium, and in battle-ships the point G is high owing to the great weights of guns and

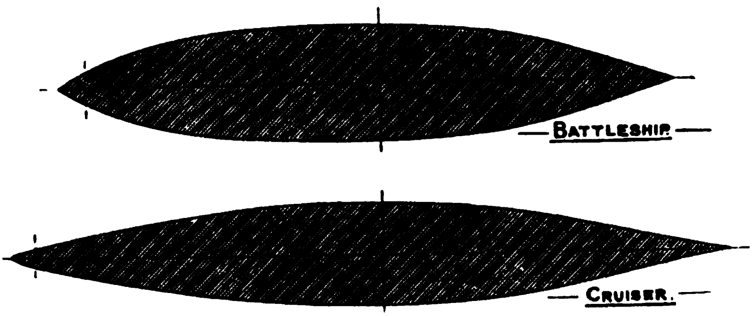


Fig. 163.—Shapes of load water plane.

armour carried high up, so that M must be high also, and this is obtained by giving such ships a large beam.

Thus *Drake*, a cruiser of 14,100 tons, carrying two 9·2-in. guns in shallow barbettes and sixteen 6-in. guns in casemates, is 71 ft. beam.

But *Duncan*, a battle-ship of 14,000 tons, carrying four 12-in. guns in massive barbettes, with twelve 6-in. guns in casemates, has to be 75½ ft. beam to obtain sufficient stability. (See Fig. 163 for comparison of shapes of waterplane of a battle-ship and a cruiser.)

A similar cause was in operation resulting in the *King Edward VII.* being made 78 ft. beam. In this ship the adoption of an armoured battery instead of casemates, and the provision of four 9·2-in. guns on the upper deck instead of 6-in. guns, caused the C.G. to be higher than in previous ships, and M had to be made higher by increasing the beam.

Examples of Stable and Unstable Equilibrium.

An example of the above principles is seen in the case of a log floating with one half its bulk immersed—say 20 ft. long and 18 in. square. It is a matter of experience that such a log will never float with one face horizontal, as Fig. 164, but always with a corner of the section downwards, as Fig. 165. The following application of the foregoing principles will explain the reason.

If the log is placed as in Fig. 164, we have the C.B. $4\frac{1}{2}$ in. from W.L., and the C.G. at half depth. To find the position of the metacentre we use the formula

$$BM = \frac{I}{V}, \text{ and } I = \tfrac{1}{12} \times 240 \times 18^3 \text{ and } V = 240 \times 18 \times 9$$

all dimensions being in inches, so that

$$BM = (\tfrac{1}{12} \times 240 \times 18^3) \div (240 \times 18 \times 9) = 3 \text{ in.}$$

but $BG = 4\frac{1}{2}$ in.

so that M is $1\frac{1}{2}$ in. below G and the log is in unstable equilibrium, and cannot float with one face horizontal.

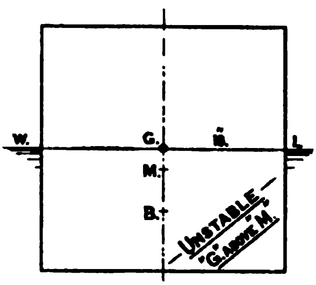


Fig. 164.

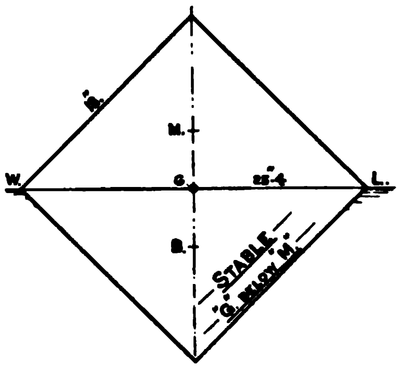


Fig. 165.

If, however, the log be placed as in Fig. 165, with one corner downwards, we have the C.B. 4·24 in. from the C.G. and

$$\begin{aligned} BM &= \frac{I}{V} \\ &= \frac{\tfrac{1}{12} \times 240 \times (18\sqrt{2})^3}{240 \times 18 \times 9} \\ &= 8{\cdot}48 \text{ in.} \end{aligned}$$

so that M is 4·24 in. above G, and the log is in stable equilibrium.

We notice that there are two influences at work in obtaining the stability in the second case, viz. (1) C.B. rises, owing to the new shape of the displacement, carrying M with it, and (2) BM is increased owing to the greater breadth at the waterplane.

Another interesting example of stable and unstable equilibrium in a floating body is seen in the case of a duck or a swan. Under ordinary circumstances

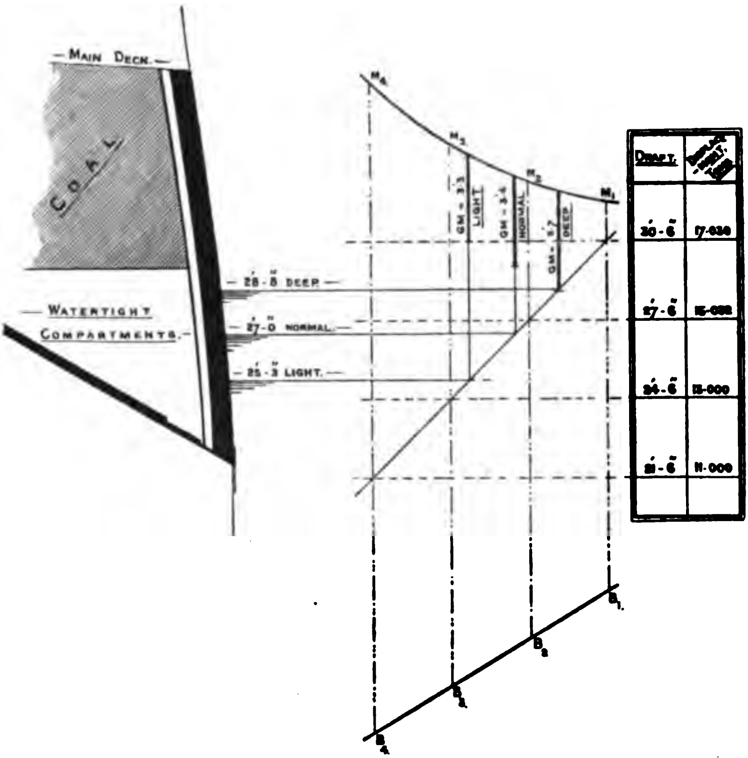


FIG. 166.—Metacentric diagram.

the shape of the waterplane is full, and the bird floats in stable equilibrium. If a swan, however, reaches down to the bottom of the water with her tail in the air, the waterplane is considerably reduced in area, being approximately circular, and this pulls down the M sufficiently to cause unstable equilibrium. In order to remain in this position it will be noticed that a swan has to work against the water with her feet to counteract the instability.

Metacentric Diagram.—The transverse metacentre, as we have seen, is a fixed point for a vessel floating at a definite waterline.

It will, however, be a different point for any other waterline, because B will shift, and the distance BM will be different. It is desirable to have some ready means of determining the position of M for any waterline at which a ship happens to be floating. This is done by constructing a *metacentric diagram*. Four or more parallel waterlines are taken, and for the ship as floating at each

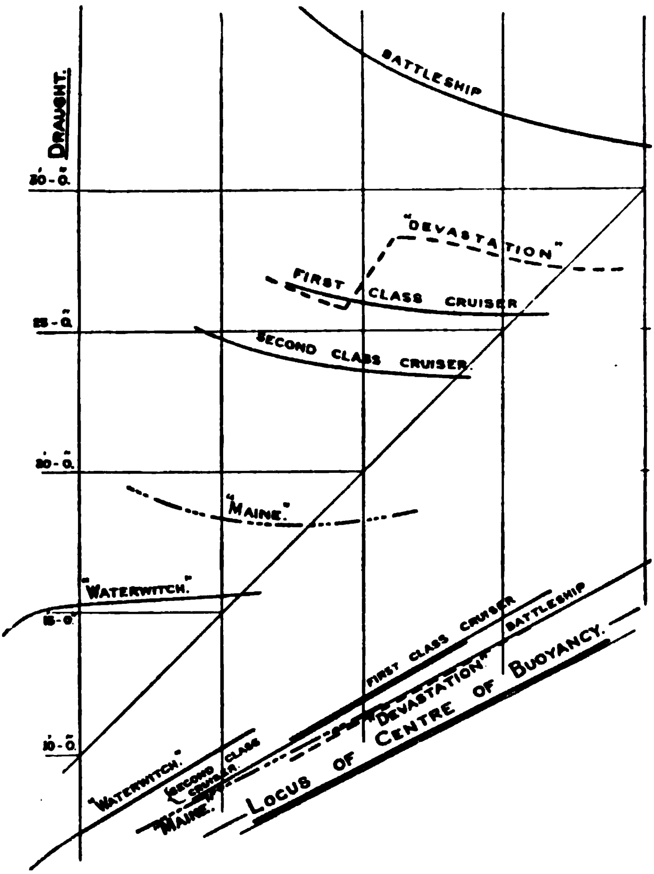


FIG. 167.

of these lines the position of the C.B. is calculated. The waterlines are set out at the proper distance apart, as Fig. 166, and a line drawn across at 45°. Through the intersections verticals are drawn and on these verticals are set down the positions of the respective centres of buoyancy, B_1, B_2, B_3, B_4. Thus for the ship shown in Fig. 166 the following were results obtained:—

At 30 ft. 6 in. waterline, C.B. is 10·35 ft. below 27 ft. 6 in. waterline
„ 27 ft. 6 in. „ C.B. is 12·00 ft. „ „ „
„ 24 ft. 6 in. „ C.B. is 13·65 ft. „ „ „
„ 21 ft. 6 in. „ C.B. is 15·3 ft. „ „ „

A curve drawn through these points gives the locus of centres of buoyancy.

In a similar manner the distance BM is determined for each of the four conditions of draught. Thus at 30 ft. 6 in. waterline, BM = 14·9 ft.; at 27 ft. 6 in. waterline, BM = 17·3 ft.; at 24 ft. 6 in. waterline, BM = 20·5 ft.; at 21 ft. 6 in. waterline, BM = 24·5 ft. These distances, set up from B_1, B_2, B_3, B_4 respectively, give the spots M_1, M_2, M_3, M_4, and the curve through these is the locus of

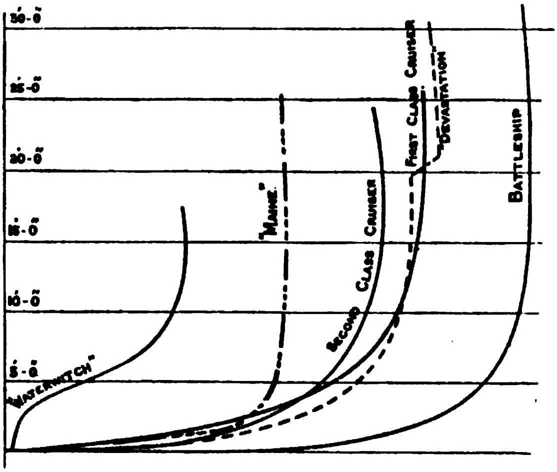


Fig. 168.

transverse metacentres. We are thus enabled to determine the position of the transverse metacentre for any waterline at which the ship happens to float. Thus at 25 ft. 3 in. draught, M is 6·4 ft. above the 27 ft. 6 in. waterline.

Fig. 167 shows the curves of C.B. and metacentres of a number of ships all placed together. The sections of these ships are shown in Fig. 168. The battle-ship is very broad, and the metacentre is consequently high. The *Devastation* has a sudden drop as she lightens, owing to the overhang of the armour. With the first and second class cruisers the finer body causes a lift in the locus of the C.B. The *Maine* is a typical merchant vessel (she has been

taken into the service, and converted into a hospital ship). She is much narrower than the other vessels, and the effect of this is seen in the low locus of metacentres. (Such ships, carrying great weights of cargo, have their C.G. low as compared with war-ships which have to carry heavy weights of guns, armour, etc., high up). The *Waterwitch* (formerly an auxiliary steam yacht, now a surveying vessel), presents some points of interest. The peg-top section causes a high C.B. As the ship lightens the M curve dips downwards, owing to the small breadth at the waterline.

Position of the Centre of Gravity.—We have been dealing above with the position of the transverse metacentre, but in order to know anything about the initial stability of a ship we must also know the vertical position of the C.G. It is possible to determine this position by means of direct calculation, and this very laborious calculation has to be done in the case of a new design. Finding the position of the C.G. in the various conditions of the ship, deep, normal, and light, it is possible to arrive at the estimated metacentric height in these conditions.

Inclining Experiment.—When a ship is completing or finished it is possible, by means of the "inclining experiment," to determine the metacentric height, and thus to find the position of the C.G. This experiment is always carried out on new ships of the Royal Navy, and also after a vessel has undergone extensive alterations likely to affect the stability. The information obtained from this experiment forms the basis of the stability calculations for the completed ship, and the main features of the vessel's stability are furnished to the Ship's Book in the *Stability Statement.* The information thus obtained also enables us to see how far the estimate of the design has been realized, and gives most valuable data for use in subsequent designs.

The main features of an inclining experiment are as follows. Weights are placed on the upper deck, as Fig. 169, and w tons, say, is moved across the deck, a distance of d feet. Then the C.G. of the ship will shift to G′, such that

$$GG' = \frac{(w \times d)}{W} \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

The ship cannot remain upright as shown, because the second condition of equilibrium is not fulfilled, viz. that the C.G. and the C.B. must be in the same vertical. The ship must heel over until the new C.B., B′, comes into the same vertical as the new

C.G., G′. Then the point where B′G′ meets the middle line is M, the transverse metacentre. If θ is the angle of heel, then

$$\tan \theta = \frac{GG'}{GM} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Substituting for GG′ from (1), we have

$$\tan \theta = \frac{w \times d}{W \times GM}$$

$$\text{or } GM = \frac{w \times d}{W \times \tan \theta} \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

The only part in this that we do not know is tan θ, and this

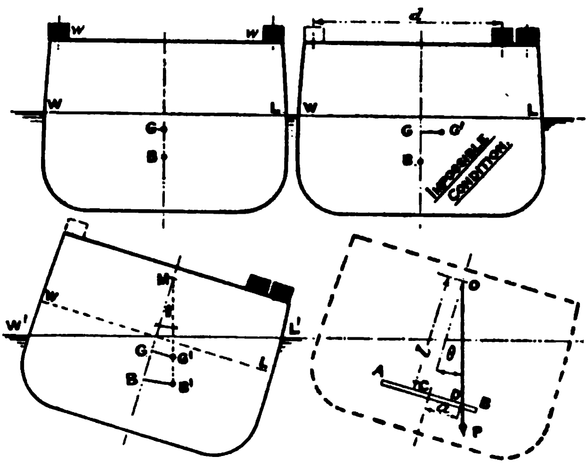


FIG. 169.—Inclining experiment.

is easily obtained by suspending two or more plumb-lines down hatchways. Then if a be the deflection along a base AB, l feet from the point of suspension, we have $\tan \theta = \frac{a}{l}$. It is desirable to have more than one pendulum, because of the check obtained on the angle. Also the weights are shifted across in various lots, both to port and starboard, so as to obtain a number of results, the mean of which can be taken.

The GM thus obtained will enable us to fix the position of G, because M will be determined from the metacentric diagram. The following is taken from an actual inclining experiment—

A vessel displacing 5372 tons is inclined by shifting 25 tons of ballast across the deck, the mean deflection observed being $10\frac{1}{4}$ in. in 15 ft.

$$\text{Here } \tan\theta = \frac{10 \cdot 25}{180}, \text{ so that—}$$

$$GM = \frac{w \times d}{W \times \tan\theta} = \frac{25 \times 36}{5372 \times \left(\frac{10 \cdot 25}{180}\right)} = 2 \cdot 9 \text{ ft.}$$

The transverse metacentre was measured off the metacentric diagram 3·1 ft. above the L.W.L. at the draught given, so that the C.G. of the ship in the inclined condition was $3 \cdot 1 - 2 \cdot 9 = 0 \cdot 2$ ft. above the L.W.L.

The ship was incomplete at the time of the experiment, so the weights to complete and remove were determined, amounting to a net total of 600 tons, with the C.G. 3·5 ft. above the L.W.L., or 3·3 ft. above the C.G. in the inclined condition. The rise of the C.G. due to this addition was—

$$\frac{600 \times 3 \cdot 3}{(5372 + 600)} = 0 \cdot 3 \text{ ft.}$$

so that the C.G. in the completed condition, 5972 tons, would be 0·5 ft. above the L.W.L. At the draught corresponding to this displacement the transverse metacentre was measured to be 2·9 ft. above L.W.L, so that the GM in the completed condition was $2 \cdot 9 - 0 \cdot 5 = 2 \cdot 4$ ft.

In this way, using the results of the inclining experiment as a basis, we are enabled to determine the draught and metacentric height in any desired condition. The usual conditions are *deep load*, *normal load*, and *light*, which have been defined in Chapter XV. These are shown on the metacentric diagram, as Fig. 166. In that case we have—

Deep load; draught 28 ft. 8 in., GM = 3·7 ft.
Normal load; draught 27 ft. 0 in., GM = 3·4 ft.
Light; draught 25 ft. 3 in., GM = 3·3 ft.

These particulars are given on the stability statement, a specimen one being given at the end of Chapter XIX.

In cruisers, in which the coal capacity is relatively very large and a large proportion of it is above the protective deck, a special condition looked into is that, supposing the upper bunkers are full, while the lower bunkers are empty. This would be an extreme condition, and the ship, unless damaged, would hardly get into a worse condition of stability than under such circumstances. If the condition thus found gives too small a GM, a special notation would be made on the stability statement giving definite instructions as to the coal stowage. In most cases, however, sufficient

stability is retained even under these extreme conditions, and a note is made on the statement that, so far as stability is concerned, the coal may be worked in any manner desired by the commanding officer.

Figs. 170 to 174 show metacentric diagrams of some typical

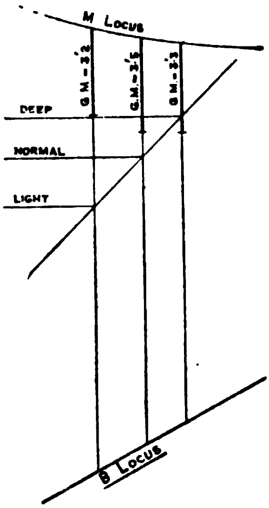


FIG. 170.

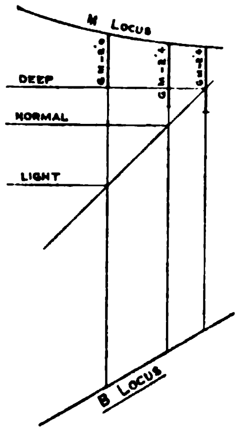


FIG. 171.

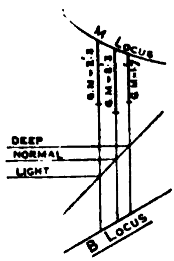


FIG. 172.

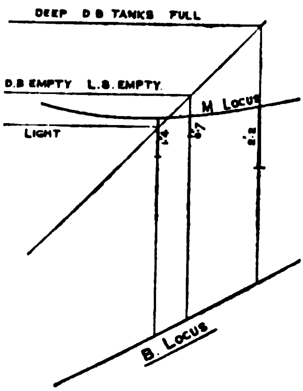


FIG. 173.

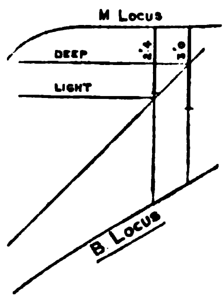


FIG. 174.

ships. We have already had the case of a battle-ship in Fig. 166. It is usually the case that, although M rises as the ship lightens to the light draught, yet G rises still more because the coal and stores

consumed are in the aggregate below the C.G. Thus the GM in the light condition is found to be less than in the other conditions. Fig. 170 is the diagram for a first class cruiser, Fig. 171 for a second class cruiser, Fig. 172 for a destroyer. The diagram for the *Maine*, originally a merchant steamer, is given in Fig. 173. In this ship it has been necessary to provide 1000 tons of permanent ballast to give proper immersion and stability, and this is so stowed so as to provide stability, when the double-bottom tanks are empty and also the lower bunkers. In this extreme condition a GM of 0·7 ft. only is obtained; but when fully equipped with double-bottom tanks full, a GM of 2·2 ft. is obtained. Fig. 174 gives the diagram for the *Waterwitch* mentioned above. In this ship, carrying considerable sail power, a good GM is required, and 3·0 ft. is obtained in the deep load condition. To get this it is necessary to stow 65 tons of permanent ballast.

Values of GM, the Metacentric Height.—The amount of GM given to a vessel is determined by the class of vessel and the qualities it is desired to obtain. To take two instances. In sailing-ships a sufficient GM must be provided to enable the ship to "stand up" under her canvas. In such a ship a small GM would mean a large angle of heel when sailing, which is undesirable. In a vessel like the *Inflexible*, in which the armour-belt only extended over one-third the length (Fig. 126), a large GM was provided, viz. 8 ft., to enable the vessel to remain upright, even supposing the unprotected ends open to the sea.

For small angles the moment of the couple tending to right a ship, *i.e.* the stability, is $W \times GM \times \sin\theta$, so that if GM is large this righting moment is large, and if GM is small, this righting moment is small. If GM is large, the ship comes back to the upright very suddenly after being inclined, and the ship will have a quick motion. Such a ship is *stiff*. If GM is small, the ship is easily inclined, but returns to the upright slowly with an easy motion.[1]

There are thus two opposing conditions to fulfil in settling the GM for a war-ship, viz.—

1. GM must be large enough to enable the ship to retain stability after a fair amount of damage.

2. GM must not be so large as to make the ship have violent

[1] This is further considered in Chapter XX. A "crank" ship easily inclined is found to be the steadiest in a seaway.

motions at sea, this being specially important in view of the fighting of guns.

The following are average values of the metacentric height given to the modern ships of the British Navy—

Battle-ships[1] ...	... $3\frac{1}{2}$ to 4 ft.
Large cruisers ...	... $2\frac{1}{2}$ to $3\frac{1}{2}$ ft.
Small cruisers ...	... 2 to $2\frac{1}{2}$ ft.
Destroyers ...	... about 2 ft.

For battle-ships it is necessary that sufficient GM shall remain after the unprotected ends are riddled and open to the sea. The gradual enlargement of the waterplane area protected during the last thirty years has been the cause of a gradual reduction of metacentric heights in these ships. Thus in the *Inflexible*, with a belt only one-third the length, a GM of 8 ft. was provided. In the Admiral class, the belt being four-ninths the length, the GM was 5 to 6 ft. For modern ships with belt about two-thirds the length, the GM is $3\frac{1}{2}$ to 4 ft. In a recent ship the effect of riddling the ends is to reduce the GM by about 2 ft. In the *Inflexible* this riddling reduced the GM by 6 ft.

Vessels like destroyers are given a GM which is relatively large. These vessels, as the speed increases, form a wave which, dipping down amidships, causes a considerable reduction of area and moment of inertia of waterplane. In consequence of this it is necessary to make the GM relatively large. Another reason for this is seen in the inward heel caused by putting the rudder over. If the GM were small, this heeling might be excessive.

For merchant steam-ships, the GM varies continually owing to the different nature and disposition of the cargo carried on different voyages. Ships with cargo of light density frequently go long voyages with metacentric heights of less than 1 ft., and their behaviour is reported to be in every way comfortable and safe. Such ships, however, have to be carefully treated when light, and frequently require water-ballast in the double bottoms to enable them to remain upright. Such a metacentric height as is sufficient in this type of ship would not be permissible in war-ships for the reasons already stated.

[1] In the *Triumph* and *Swiftsure*, recently added to the Royal Navy, the metacentric heights are—

Deep load ...	... 3·75 ft.
Normal load ...	... 3·5 ft.
Light	... 2·7 ft.

Sailing ships are given metacentric heights of 3 to 3½ ft. to enable them to "stand up" under their canvas without heeling to an undesirable angle.

Heel caused by Flooding the Wings, etc.—The following figures are interesting as showing the heel caused by opening up one side of a battle-ship to the sea, in way of the engine-room:—

1. Wings only open to the sea (no coal in). A heel of 5° would result.

2. Wings and inner bunkers (no coal in) open to the sea. A heel of 10° would result.

3. Wings, bunkers (no coal in), and one engine-room open to the sea. A heel of 18° would result.

In either of these cases it would be advisable to admit water to the wings on the opposite side of the ship, to restore the vessel to the upright. Not only because the guns on the opposite side to the damage would be unable to get horizontal fire, but because the lower edge of armour comes out of water between 8 and 10°, leaving the vitals of the ship completely unprotected (see Fig. 130). In such ships provision is made for flooding the wings, if necessary, to correct heel or trim. We have already seen that the middle line bulkhead of the engine-room is made strong enough to stand the pressure, supposing one engine-room to be flooded.

A similar state of things obtains in vessels with a middle line bulkhead through the boiler-rooms, like the *Majestic*. In more recent vessels, however, this bulkhead is not fitted, because of the new arrangement of watertube boilers. The flooding of a boiler-room, therefore, causes a bodily sinkage; the flooding of wings, etc., will cause heeling, as seen above.

Influence of Coal stowed in Upper Bunkers.—1. The question of coal in the upper bunkers at the side of war-ships is important because of the resistance such coal offers to direct penetration. It has been found that 2 ft. of coal is equivalent in resisting power to 1 in. of iron. This is specially important in deck-protected cruisers (Figs. 21, 22, 24, 26), which depend so largely on the coal above the protective deck for their protection, and on this account the coal in the upper bunkers at the side should be the last to be used. All ships of this type in the Royal Navy have sufficient stability even supposing all the coal in the lower bunkers burnt out and the upper bunkers completely full. In the sloops, which are quite unprotected save

by the coal, a division is placed in the bunkers, so that some coal may remain above the flat as long as possible, in order to retain its protection (Figs. 29, 30).

2. In addition to this direct protection there is the fact that the bodily sinkage on riddling the side and admitting water would be less with coal than without it. Every cubic foot of bunker space, with coal in, contains five-eighths solid space occupied by the coal, and three-eighths vacant space, into which water could penetrate. The influence of this in limiting bodily sinkage has already been seen in Chapter XVI.

3. Of greater importance, however, than (1) or (2) is the influence of the coal in preserving the initial stability, supposing the side of the ship is riddled in way of the upper bunkers.

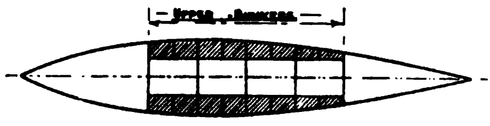


FIG. 175.

(*a*) With the side of the ship intact, the shaded parts in Fig. 175 will contribute their full value to the moment of inertia of the waterplane.

(*b*) With the side riddled and no coal, the shaded parts will contribute nothing to the moment of inertia of the waterplane.

(*c*) With the side riddled and coal in, the shaded parts will contribute five-eighths their area to the moment of inertia of the waterplane.

We have seen that the position of the transverse metacentre, on which the initial stability so largely depends, is directly influenced by the transverse moment of inertia of the waterplane.

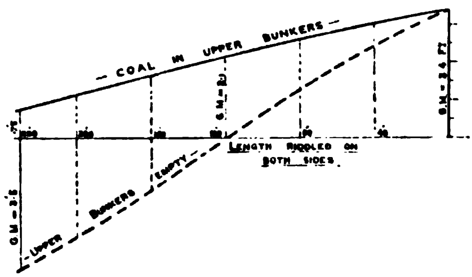


FIG. 176.

Fig. 176 gives the result of calculations made on a cruiser 480 ft. × 71 ft. × 26 ft. × 13,000 tons, in which the sides in way of bunkers on *both sides of the ship* were assumed to be gradually riddled. The vessel starts with a GM of 3·4 ft. If coal is in the upper bunkers the riddling of both sides for the

length of 230 ft. leaves the ship with a GM of 9 in., so that the ship although tender would be stable. If the bunkers are empty the reduction of GM as the sides are riddled is much more rapid, and the vessel would become unstable when about half the length of bunkers was riddled.

Fig. 177 gives the result of calculations made on a cruiser 370 ft. × 57 ft. × 20 ft. × 6160 tons, having *one side* only gradually riddled in way of the upper bunkers, and the consequent heel corrected by the admission of water to the wing compartments or double bottom respectively. The curve AA shows that, with coal in, the riddling of the side for the whole bunker length leaves the ship with a GM of 2½ ft., supposing that water is admitted to the wings to keep her upright. BB is the curve, supposing the coal out of the upper bunkers and the heel corrected by admitting water to the wings. The ship still retains a GM of 1 ft., supposing the length of 165 ft. is riddled. A rather smaller GM is found to result if the heel is corrected by admitting water to the double bottoms only, curve CC. In these cases the lowering of M due to the loss of moment of inertia of the waterplane is partially compensated for by the lowering of G due to the admission of water to the wings or double bottoms.

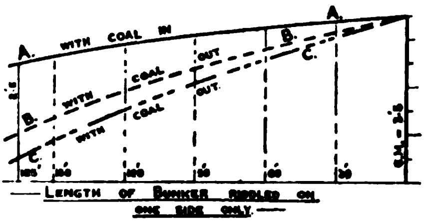


FIG. 177.

The following example is introduced to show that a given quantity of coal on board a vessel is more usefully disposed, as regards stability, when the side is riddled, when in the upper bunkers at the side than in the lower bunkers, in spite of the lowering of the C.G. of the ship that takes place when the coal is trimmed down.

EXAMPLE.—*A box-shaped vessel is* 350 *ft.* × 60 *ft. with bunkers at the side amidships* 10 *ft. wide,* 160 *ft. long, extending from* 14 *ft. to* 26 *ft. above the keel. When these bunkers are full the draught is* 20 *ft., and the metacentric height* 3 *ft. Determine the effect on the initial stability—*

(i.) *With sides riddled in way of upper bunkers, coal in.*

(ii.) *With sides riddled in way of upper bunkers, the coal having been trimmed to the lower bunkers* 8 *ft. above keel.*

Taking 43 cubic ft. of coal to the ton, the side bunkers will hold about 900 tons. The displacement of vessel is 12,000 tons.

(i.) With the sides intact M is 25 ft. above keel, and G consequently 22 ft. above keel. In the bunkers five-eighths the space is occupied by the coal, and three-eighths the space is vacant. When the sides are riddled the vessel sinks slightly, and as the moment of inertia of the waterplane is reduced by the admission of water to the space in the bunkers unoccupied by the coal, from its original value of 6,300,000 to 5,540,000. This causes M to drop to 23·2 ft. from keel, and the resulting metacentric height is 1·2 ft. The vessel is therefore stable since M is above G.

(ii.) If now we trim the coal to the lower bunkers, we depress the C.G. $\frac{(900 \times 12)}{12,000} = 0{\cdot}9$ ft., so that the GM is 3·9 ft., and G is 21·1 ft. above keel.

On riddling the sides we get a greater sinkage, but the chief effect is the reduction of the moment of inertia of the waterplane to 4,274,000, the area of the waterplane in the bunkers contributing nothing to the moment of inertia. This gives the point M 20·36 ft. above keel, or 0·74 ft. below the C.G. The vessel is thus unstable in the upright condition.

Methods of increasing the Metacentric Height of a Vessel.—1. If we put ballast, either pig-iron or water, into the lower part of the ship, there will be two effects, viz.—

(*a*) Increase of draught, and

(*b*) Depression of the C.G. of the ship.

The increase of draught, if moderate, will not in general alter the position of the transverse metacentre very much; whether it does so or not will depend on the shape of the metacentric diagram. If w be the added weight, W the weight of the ship before the addition, and d the distance of the added weight below the C.G., then

$$\text{depression of C.G.} = \frac{w \times d}{(W + w)}$$

Thus the addition of the weight of 20 tons, 10 ft. below the C.G. of a vessel of 1000 tons displacement, will lower the C.G. of the ship $\frac{20 \times 10}{1020} = 0{\cdot}2$ ft. In general the GM will increase by this amount, but if the M locus in the metacentric diagram slopes down sharply as the draught increases the increase of GM may be rather less than thus obtained.

In adding water-ballast to the double bottom of a vessel it is essential that each compartment should be completely filled so that the water will act as a solid weight. If a free surface is left the water can shift over to the side to which the ship is heeling, and this tends to counteract the increased GM obtained by the water-ballast. The manholes to double bottoms are always made

with raised coamings, in order to ensure the compartments being completely filled (see Fig. 48). Air escapes are also provided.

There may be cases where it is undesirable to increase the draught of a ship by adding ballast, and yet it is necessary to obtain greater initial stability. In such cases the following method, or No. 3 below, would have to be adopted.

2. If top weight is taken out of a ship there will be two effects, viz. (1) decrease of draught, and (2) depression of the C.G. Thus a ship is of 5000 tons displacement. The effect of removing two military tops weighing 24 tons, originally 70 ft. above the C.G., would be to cause depression of the C.G. $\frac{(24 \times 70)}{(5000 - 24)} = 0{\cdot}34$ ft., and this will be generally the increase of metacentric height, unless there is something exceptional about the metacentric diagram.

3. The two previous methods were concerned with lowering the C.G., the present method deals with the metacentre. It will be remembered that the position of the metacentre is directly dependent on the moment of inertia of the waterplane. If we can increase this we raise the metacentre, and so increase the initial stability. This can be done by placing a girdling at the waterline, over the midship portion of the length, as Fig. 178. This adds very little to the draught, but considerably to the moment of inertia of the waterplane. This method of increasing the stiffness used to be frequently adopted in the wooden sailing-ships in order to enable them to "stand up" better.

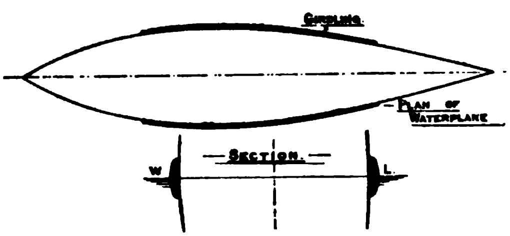


FIG. 178.

An instance of its adoption in the Royal Navy was in the case of the *Sultan*. This ship had to undergo an extensive reconstruction, and it was found that the alterations would leave her with insufficient stability. The best way to increase the stability was found to be by adding a wooden girdling over the midship portion of the length, as the addition of any weight on board was undesirable.

Stability when partially Waterborne.—An application of the

principles of the present chapter, of interest and some importance, is seen in the reduction of stability which takes place when a vessel is partially waterborne. This happens when a vessel is being docked or undocked, and also if a vessel is run on to a shelving beach. In Fig. 179, suppose a ship is being docked, and the water level falls from W′L′ to W″L″. If we suppose a small inclination θ, the support of the displacement of the zone between W′L′ and W″L″, viz. w, which originally acted through b, the C.G. of the zone, now acts at the keel, and the buoyancy $W - w$ acts in

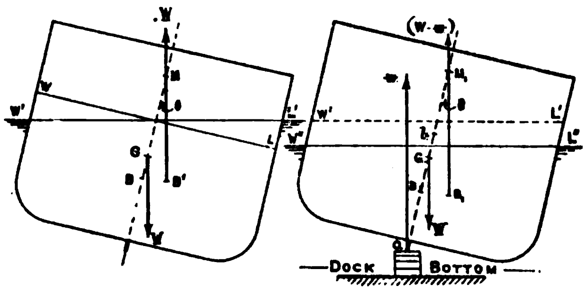


FIG. 179.

the line B_1M_1, where M_1 is the metacentre corresponding to the waterline W″L″.

The original moment righting the ship was $W \times GM \times \sin \theta$, but the moment now righting the ship is—

$$\begin{aligned} &\{(W - w) GM_1 - w.OG\} \sin \theta \\ = &\{(W \times GM_1) - (w \times OM_1)\} \sin \theta \end{aligned}$$

since the influence of w is to upset the ship.

It may be shown that the reduction of metacentric height thus caused is $\left(\frac{w}{W} \times Ob\right)$.

In the case of a ship being docked, the critical point is reached when the keel is just taking the blocks all fore and aft, and the time until this happens is longer in the case of a ship trimming a great deal by the stern than in a ship on a more even keel. In such a ship, therefore, the support w may reach a considerable amount before the ship takes the blocks, after which the shores can be set up. Just before the shores are set up, there is, therefore, a reduction of stability which may be sufficient to render a

ship unstable. It is necessary, therefore, when docking and undocking to keep the ship well under control to prevent any transverse inclinations while any of the weight is taken by the blocks.

For ordinary ships the loss of metacentric height thus caused will not be sufficient to reduce the GM enough to cause instability, but it is possible in a ship having large trim and small metacentric height when being docked.

It is important to note in connection with the docking of ships that a ship with small GM should never be undocked, if, while in dry dock, any alteration of the weights on board is made which tends to reduce the metacentric height, unless other weights are added to compensate. For example, a merchant ship when light may require water-ballast to keep her upright. If docked in this condition the ballast must not be removed while in dock (unless compensation is made), or else it would be found that when the ship was again afloat she would be unstable.

CHAPTER XVIII.

TRIM, MOMENT TO CHANGE TRIM ONE INCH, ETC.

We have now to deal with inclinations in a fore-and-aft or longitudinal direction. As the stability of a ship is a *minimum* for transverse inclinations, so the stability is a *maximum* for longitudinal inclinations. We do not need, therefore, to study the

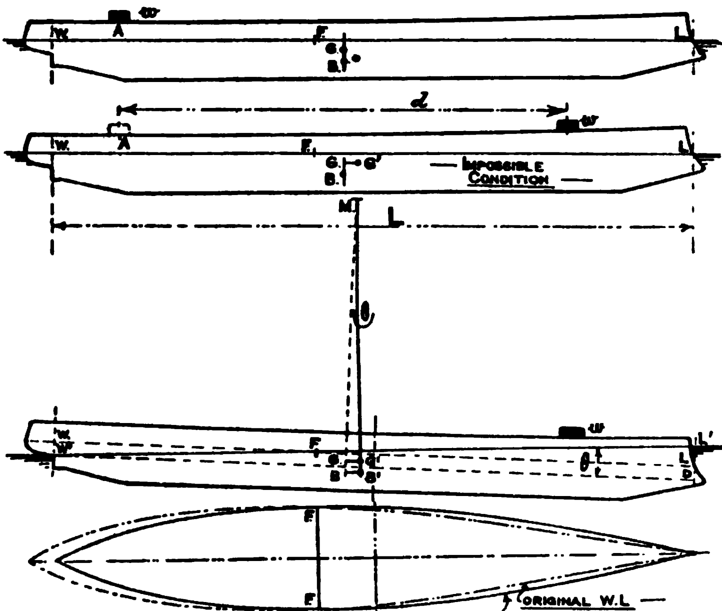


Fig. 180.

longitudinal stability of a ship to ascertain whether she is safe or not, as we do the transverse stability, but in order to deal with questions of trim or forward and after draughts.

If a ship, Fig. 180, is floating originally at a waterline WL,

and by some means is made to float at the waterline W'L', the centre of buoyancy must shift, owing to the changed shape of the displacement, from B to B' say. Then the original vertical through B and the new vertical through B' will intersect in the point M, which is the *longitudinal metacentre.* This point is precisely analogous to the transverse metacentre, the difference consisting in the direction of the inclination. The distance between the C.G. and the longitudinal metacentre is the *longitudinal metacentric height.*

The point M is determined with reference to the C.B., and the distance BM is given by the equation—

$$BM = \frac{I_0}{V}$$

where I_0 is the moment of inertia of the waterplane about a transverse axis through its centre of gravity (this C.G. is termed the *centre of flotation*), and V is the volume of displacement.

The calculation for I_0 is somewhat complicated,* but it may be approximately written—

$$I_0 = n'.L^3.B \text{ (n' being a coefficient)}$$

$$\text{also } V = k.L.B.D \text{ (k being the coefficient of fineness)}$$

$$\text{so that } BM = \frac{n'}{k}\cdot\frac{L^2}{D} = b\cdot\frac{L^2}{D} \text{ approximately}$$

where L is the length of ship between perpendiculars in feet
D is the mean draught in feet
b is a coefficient which does not vary much from 0·075.†

This approximate formula shows the great influence of the length in determining the position of the longitudinal metacentre.

Longitudinal shift of Weights already on Board.—The *trim* of a ship is the difference between the forward and after draughts. Thus H.M.S. *Pelorus* is designed, under normal load conditions, to float at a draught of 12 ft. forward and 15 ft. aft, giving a *trim* of 3 ft. by the stern.

Change of trim is the sum of the changes of draught forward

* See the author's "Theoretical Naval Architecture."

† In a vessel with full waterplane n' will be large, but k will also be large, the ship being of full form. If n' is small, k also will be small. So that the quotient $\frac{n'}{k}$ does not vary much for ordinary ships.

and aft. Thus if *Pelorus*, when floating at 12 ft. forward, 15 ft. aft, has certain weights shifted on board resulting in a draught of 12 ft. 10 in. forward and 14 ft. 2 in. aft, she is said to have changed trim 10 in. + 10 in. = 20 in. Change of trim can be produced by the fore-and-aft shift of weights already on board, this being analogous to the inclining experiment, in which heeling is caused by shifting weights in a transverse direction.

In Fig. 180, let w be a weight on the deck at A when the vessel is floating at the waterline WL. Now suppose w is moved forward through a distance d. G, the centre of gravity of the ship, will in consequence move parallel to the shift of w to G′, such that $GG' = \frac{(w \times d)}{W}$. Under these circumstances the vessel cannot float at the waterline WL, as in the second sketch in Fig. 180, because the C.G. and the C.B. are not in the same vertical line. The ship must adjust herself to the line W′L′, as shown, so that G′, the new centre of gravity, and B′, the new centre of buoyancy, are in the same vertical. Then the line through B′G′ intersects the original vertical through BG, in M, the longitudinal metacentre.

If θ is the small angle of inclination—

$$\tan \theta = \frac{GG'}{GM}$$

but also—

$$\tan \theta = \frac{DL'}{\text{length}} = \frac{WW' + LL'}{\text{length}} = \frac{\text{change of trim}}{\text{length}}$$

Having thus two values of $\tan \theta$ we can equate, so that—

$$\frac{\text{change of trim}}{\text{length}} = \frac{GG'}{GM} = \frac{w \times d}{W \times GM}$$

using the value found above for GG′.

We therefore have—

$$\textit{change of trim in feet} = \frac{w \times d}{W} \times \frac{L}{GM}$$

$$\text{and the } \textit{change of trim in inches} = \frac{w \times d}{W} \times \frac{L}{GM} \times 12$$

Transposing we have the moment causing the change of trim, viz.—

$$w \times d = \frac{W \times GM}{12 \times L} \times \text{change of trim in inches}$$

and if the change of trim be 1 in.—

$$\textit{Moment to change trim 1 in.} = \frac{W \times GM}{12 \times L} \text{ foot tons}$$

where W = displacement in tons
GM = longitudinal metacentric height
L = length between draught marks

In ships of small ratio of length to breadth, like battle-ships up to *Royal Sovereign,* the longitudinal GM was approximately equal to the length. In these ships, therefore, the moment to change trim 1 in. is very nearly one-twelfth the displacement in tons. This does not hold so well for more modern ships, and the following give good approximations to this moment to change trim 1 in. :—

For vessels of full form like battle-ships $\frac{1}{9{,}000} \times L^2 \times B.$

For vessels of finer form like cruisers $\frac{1}{11{,}000} \times L^2 \times B.$

where L = length between perpendiculars in feet
B = breadth of ship in feet

The length and breadth do not vary much for considerable changes of draught; the above formulæ, therefore, show that the moment to change the trim 1 in. varies very little for the range of draughts at which a vessel is likely to float.

The following examples will illustrate the use of the above in determining change of trim :—

EXAMPLE.—*A vessel is floating at a draught of* 12 *ft.* 1 *in. forward and* 14 *ft.* 10 *in. aft. Determine the draughts forward and aft after shifting* 5 *tons from forward to aft through* 210 *ft. The moment to change trim* 1 *in. is* 295 *foot tons.*

In this case $w = 5$ tons, $d = 210$ ft., so that $w \times d = 1050$ foot tons. The change of trim is therefore $\frac{1050}{295} = 3\frac{1}{2}$ in. This change of trim will cause an increase of draught aft of $1\frac{3}{4}$ in., and a decrease of draught forward of $1\frac{3}{4}$ in. So that—

draught forward = 12 ft. 1 in. − $1\frac{3}{4}$ in. = 11 ft. $11\frac{1}{4}$ in.
draught aft = 14 ft. 10 in. + $1\frac{3}{4}$ in. = 14 ft. $11\frac{3}{4}$ in.

EXAMPLE.—*Determine approximately the shift of* 50 *tons on board the above ship necessary to bring her to an even keel.*

The mean draught is—

$$\frac{12 \text{ ft. } 1 \text{ in.} + 14 \text{ ft. } 10 \text{ in.}}{2} = 13 \text{ ft. } 5\tfrac{1}{2} \text{ in.}$$

There is therefore necessary a change of trim of—

$$(13 \text{ ft. } 5\tfrac{1}{2} \text{ in.} - 12 \text{ ft. } 1 \text{ in.}) + (14 \text{ ft. } 10 \text{ in.} - 13 \text{ ft. } 5\tfrac{1}{2} \text{ in.}) = 33 \text{ in.}$$

The necessary moment from aft forward is therefore $295 \times 33 = 9735$ foot tons, so that 50 tons would need to be shifted $\frac{9735}{50} = 195$ ft., say, from aft forward in order to bring the ship to an even keel.

Effect on the Trim due to adding a Weight of Moderate Amount.—If a ship sinks from a waterline WL to a parallel waterline W′L′, the added buoyancy of the layer will act at the C.G. of the layer. If the sinkage is small this point will very nearly be in the same section as the centre of flotation. If, therefore, we wish to add a weight to a ship so that the trim shall not be changed, the added weight must be in the same section as the added buoyancy, and if the added weight be not large, it must, therefore, be placed in the same section as the centre of flotation. The centre of flotation in ships of the Navy is usually abaft amidships, on the average about one twenty-fifth the length.

If a weight is placed on board anywhere else both increase of draught and change of trim must occur. We imagine the weight is added first at the centre of flotation, by which we can see how much she will sink to a parallel waterline, and then we shift the weight to the position given, and determine the change of trim. The following will illustrate the method of dealing with the addition of weights on board a ship.

EXAMPLE.—*A ship is floating at a draught of* 20 *ft. forward and* 22 *ft. aft, when the following weights are placed on board in the positions given, viz.*—

20 *tons,*	100 *ft.*	*before*	*the centre*	*of flotation*
45 ,,	80 ,,	,,	,,	,,
60 ,,	50 ,,	*abaft*	,,	,,
30 ,,	10 ,,	,,	,,	,,

What will be the new draught, the moment to change trim 1 *in. being* 800 *foot tons and the tons per inch* 35.

The total weight added is 155 tons, and if placed at the centre of flotation, the increased draught is $\frac{155}{35} = 4\frac{1}{2}$ in.

The weights to be moved forward give a moment of $(20 \times 100) + (45 \times 80) = 5600$ foot tons, and the weights to be moved aft give a moment of $(60 \times 50) + (30 \times 10) = 3300$ foot tons. The forward moment is thus in excess by 2300 foot tons, and this will cause a change of trim of $\frac{2300}{800} = 3$ in. nearly, and the new draught forward is 20 ft. $4\frac{1}{2}$ in. + $1\frac{1}{2}$ in. = 20 ft. 6 in., and the new draught aft is 22 ft. $4\frac{1}{2}$ in. − $1\frac{1}{2}$ in. = 22 ft. 3 in.

Strictly speaking, the change of trim ought not to be divided equally forward and aft. It should be less than half (about 0·46) aft, and rather more than half (about 0·54) forward. This will be understood by reference to Fig. 180. Since F, the centre of flotation, is abaft amidships, WW′ is not equal to LL′, but rather less. Unless the change of trim is very considerable, however, the error involved in taking the half forward and aft is small.

The following example will illustrate how to deal with the problem of bringing a vessel to such a draught as will allow her to pass over a place like a bar at the mouth of a river.[1] It is assumed also that no information is available, except the dimensions of the ship, so that the approximations already given have to be employed.

EXAMPLE.—*A cruiser 300 ft. × 36½ ft. is floating at a draught of 12 ft. forward and 15 ft. aft. It is desired to bring her to an even keel at a draught of 12 ft., in order to pass over a bar at the mouth of a river. Determine approximately how this could be accomplished.*

The mean draught is 13 ft. 6 in., so that the ship would need lightening 18 in. The approximate tons per inch (page 174) would be $\frac{300 \times 36 \cdot 5}{600} = 18 \cdot 25$ tons, so that $18 \times 18 \cdot 25 = 330$ tons, say, must be removed. Suppose this is done so that the vessel lightens to a draught of 10 ft. 6 in. forward, 13 ft. 6 in. aft.

The approximate moment to change trim 1 in. is $\frac{1}{11000} \times (300)^2 \times 36 \cdot 5 = 300$ foot tons, say. As a change of trim of 36 ins. is necessary to bring the vessel to an even keel, we require a moment of $300 \times 36 = 10{,}800$ foot tons. The 330 tons must therefore be removed $\frac{10800}{330} = 33$ ft., say, abaft the centre of flotation to give the necessary moment. Taking the centre of flotation as $\frac{1}{25}$ the length abaft amidships, or 12 ft., *the 330 tons would have to be removed so that its C.G. was about 45 ft. abaft amidships to give the required draught of 12 ft. forward and 12 ft. aft.*

(It may be stated that, taking the correct data for the above ship, the answer would be 350 tons, taken out 42½ ft. abaft amidships, so that the approximation is a very good one.)

Passage through the Suez Canal.—In order to pass through the Suez Canal it is necessary that the maximum draught should not exceed 26 ft. 3 in. In the deep draught battle-ships like *Majestic* class which may have to pass through the canal, a special set of instructions is drawn up, showing how the ship may most readily be brought to the required mean draught. The following is a specimen set of such instructions (the maximum draught allowable has since then been increased to 26 ft. 3 in., instead of 25 ft. 7 in.).

[1] If, as would probably be the case, the bar is in fresh water, allowance must be made for the bodily sinkage in going from salt to fresh water, see p. 177.

In order that the vessel may pass through the Suez Canal, the extreme draught should not exceed 25 ft. 7 in., which may be obtained in the following manner:—

1. Remove all water from the hydraulic tanks, the after fresh-water tanks, and all except 20 tons from fresh-water tanks forward.

2. Remove all water from boilers in the after boiler-rooms; the water in the other boilers should be to working height.

3. Empty the reserve feed-tanks, with the exception of about 10 tons of water.

4. Remove all coal except 200 tons in bunkers abreast the four forward boilers.

5. Remove all provisions, bread, and spirits, except 20 tons; the provisions, etc., removed should be principally from the after store-rooms.

6. Remove all officers' stores and slops except 30 tons, removing principally from the after store-rooms.

7. Remove 55 tons of shell from forward 12-in. shell-room, and an equal quantity from the after 12-in. shell-room; also 40 tons of shell from forward 6-in. shell-room, and an equal quantity from the after 6-in. shell-room.

Generally.

1. When the ship is floating at or near a mean draught of 27 ft., a weight of 675 tons added to or removed from the ship will increase or decrease the mean draught by 1 ft.

2. A longitudinal moment of about 15,800 foot tons will alter trim by 1 ft., *i.e.* if w tons is weight moved, and d feet the distance the weight is moved longitudinally, then $\frac{(w \times d)}{15,800}$ is *the change of trim in feet*, or the increase of draught aft plus the decrease of draught forward, or *vice versâ.*

3. A weight placed on board or removed from about station 93, which is about the middle of the after boiler-room, will not affect the *trim.*

4. To ascertain the combined effect on draught and trim of removing a weight or placing a weight on board, the effect on draught only is first obtained by rule (1) above, supposing the weight put on board at station 93; the effect on the trim is then obtained by rule (2) above, the distance moved through being the distance in feet between station 93 and the actual position of the weight.

Note.—The removal of any weight before station 68, which is 12 ft. abaft the forward bulkhead of forward boiler-room, will not diminish the draught aft.

Similarly, any weight removed from abaft station 116, which is abreast the mainmast, will not decrease the draught forward.

It should be stated that more recent battle-ships and cruisers have been designed to float at a rather less draught with legend coal, etc., than the ships of *Majestic* class, and this, together with the increase of draught now allowable, renders the operation of lightening these ships to pass through the canal considerably simpler than the above.

Stability of a Submarine Boat.—In a vessel totally submerged the shape of the displacement does not alter for any inclination (Fig. 181), and therefore the upward force of the buoyancy must always act through the same point, viz. the centre of buoyancy. The stability at any angle θ is W $\times$ BG $\times$ sin θ, varying directly as sin θ. It will be a maximum at 90°, where sin θ is a maximum, and the angle of vanishing stability[1] will be 180°, where sin $\theta = 0$. In order to give good stability, therefore, BG must be as large as possible. This is done by so arranging the weights and ballast that G is below B.

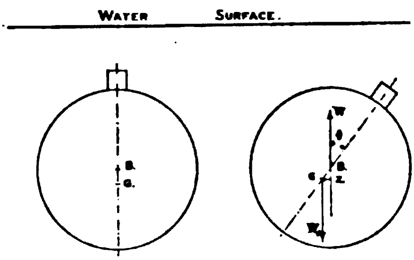


FIG. 181.—Stability of a submarine.

For longitudinal inclinations the distance between B and G is the same as for transverse inclinations, so that the case is very different from an ordinary ship, in which the stability for fore-and-aft inclinations is very great. In a submarine, therefore, the stability is the same for all directions of inclination, and such a boat is exceedingly sensitive to anything tending to disturb the fore-and-aft position.

Change of Trim after Bilging.—When a vessel is bilged near either end both bodily sinkage and change of trim occur.

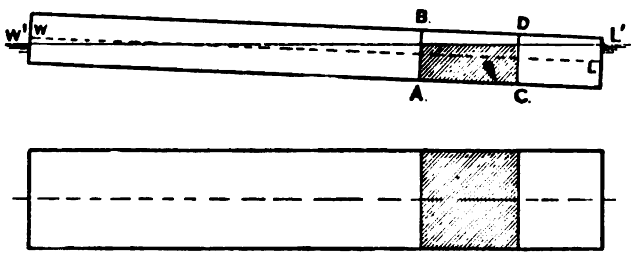


FIG. 182.

This is well illustrated by the case of the *Victoria* in Chapter XXIV. In that case the change of trim was so considerable as to bring the upper deck forward under water, and consequently water gained

[1] See next chapter.

access to the ship through hatchways, etc., and the movement down by the head was greatly accelerated. Fig. 182 has been drawn for a box-shaped vessel 175 ft. long, 30 ft. broad, 15 ft. deep, 8 ft. draught, before damage. If an empty compartment between bulkheads 25 ft. and 55 ft. from the bow is laid open to the sea the vessel will float at a draught of 13 ft. 5 in. forward and 6 ft. 8 in. aft. It is seen that the stem head is quite close to the water, and although the loss of buoyancy is not very considerable, yet this, with the change of trim, causes a dangerous condition. It is thus seen to be most important to carry watertight transverse bulkheads well above water. Figs. 52 and 54, which show the watertight subdivision of a large and small cruiser respectively, show that most of these bulkheads are carried to the upper deck.

CHAPTER XIX.

STABILITY AT LARGE ANGLES OF INCLINATION.

We have seen that the stability of a ship at any angle is the effort she makes to return to the upright when put over to that angle. For small angles of inclination, up to 10 to 15°, this depends directly on the metacentric height. Thus at 10° the *Royal Sovereign*, with $3\frac{1}{2}$ ft. GM and 14,150 tons displacement, will have a righting moment of

$$14{,}150 \times 3{\cdot}5 \times \sin 10° = 8{,}600 \text{ foot tons.}$$

It is possible, however, for a vessel to have sufficient metacentric height but insufficient stability at large angles. This was specially brought out in the investigations which followed the loss of H.M.S. *Captain*. Metacentric height alone, apart from other considerations, principally freeboard, will not ensure a vessel having sufficient stability, and special calculations are necessary to determine the righting moment at large angles of inclination.

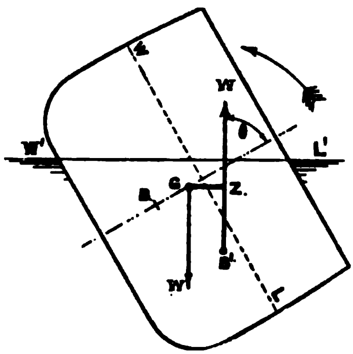


FIG. 183.

Curve of Stability.—Take a vessel inclined to a large angle θ, as Fig. 183. The upward force of the buoyancy acts through B′, the new centre of buoyancy, and the couple tending to right the ship is W × GZ, GZ being the *righting lever*. The length of this righting lever will depend on how far the centre of buoyancy shifts out, and this length will vary for different angles. Thus for H.M.S. *Captain* the following values of GZ were calculated, viz. 7°, $4\frac{1}{4}$ in.; 14°, $8\frac{1}{2}$ in.; 21°, $10\frac{3}{4}$ in.; 28°, 10 in.; 35°, $7\frac{3}{4}$ in.; 42°,

$5\frac{1}{4}$ in.; $54\frac{1}{2}°$, zero. A convenient way of representing these results is to draw a base line to represent angles of inclination and set up as ordinates the lengths of GZ as found. A curve drawn through the spots thus obtained is a *curve of statical stability.* The curve for the *Captain* is in Fig. 184.

Fig. 185 shows a curve of stability constructed as above. The angle at which GZ obtains its maximum value is termed the *angle of maximum stability* (in this case 47°). The angle at which the

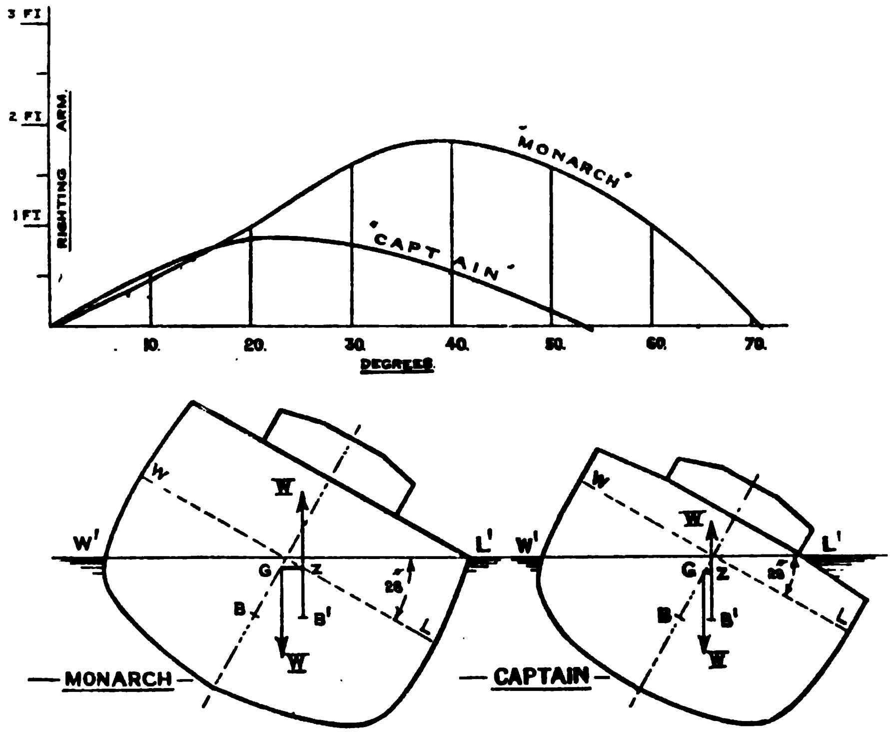


FIG. 184.

curve crosses the base line (in this case 77°) is termed the *angle of vanishing stability,* or the *range of stability.* Up to this angle the vessel possesses a righting-lever which will take her back to the upright. At 77° the ship is in *equilibrium,* the C.G. and C.B. being in the same vertical, but this equilibrium is *unstable,* and a small inclination either side of 77° will take her away from that angle; if to 75°, say, she will go back to the upright; if to 79°, say, she will capsize.

In striking contrast to the curve for the *Captain* is that for the *Monarch* (Fig. 184). In this case the angle of maximum stability is not reached until 40° as against 21° in the *Captain*, and the value of the maximum GZ is about twice as great. The stability does not vanish until the large angle of 70° is reached. The reason for the difference between the two ships is seen by comparing the sections. The *Monarch* had high freeboard, which pulls out the centre of buoyancy as the ship heels over. The *Captain* had a low freeboard, giving a curve of stability which was dangerously small for a ship carrying a large amount of sail.

In considering a curve of stability certain assumptions have to

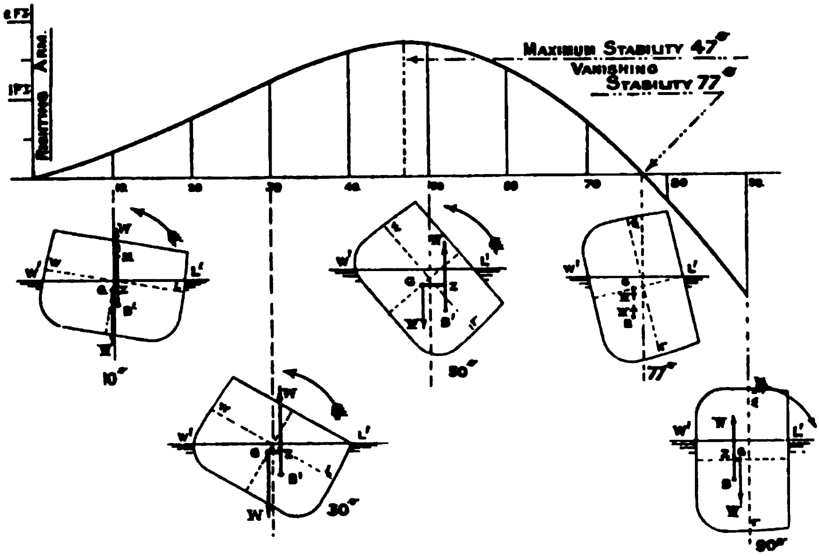


Fig. 185.

be made, which are necessary, in order to make the calculations at all feasible.

1. The sides and deck are assumed to be watertight for the range over which the curve is drawn. Thus all sidelights and gun-ports are supposed to be closed below the upper deck. If the effect of the forecastle or poop is included, any openings in these superstructures are supposed to be closed.

2. The C.G. of the ship is taken in the same position in the ship throughout the inclination, *i.e.* it is supposed that no shift of weights takes place.

The important features of a curve of stability are:—

1. Inclination which the curve has to the base line at the start. This inclination depends directly on the metacentric height.

2. The angle at which the maximum value of the righting arm occurs and its value at that angle.

3. The range or the angle at which the curve crosses the base line, and the vessel becomes unstable.

Effect on a Curve of Stability by variation of Beam, Freeboard, and Position of C.G.—In order to illustrate these points a box-shaped vessel is taken having a breadth of 50½ ft., draught of 21 ft., freeboard of 6½ ft., and metacentric height of 2·6 feet.

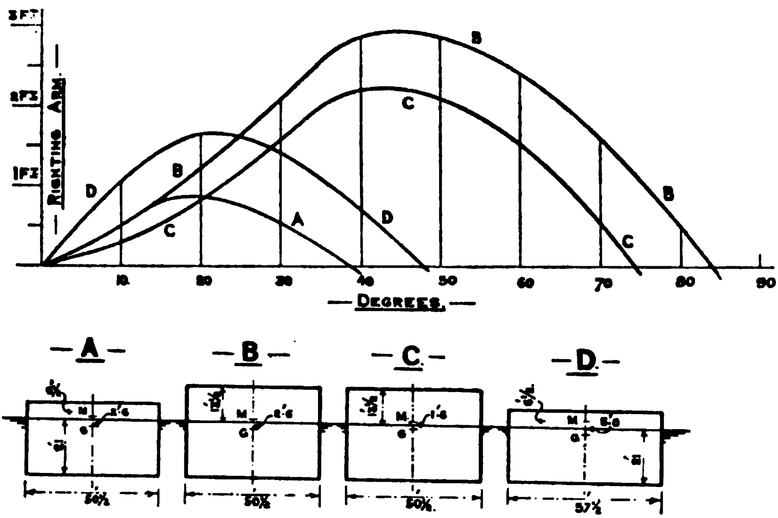


FIG. 186.

The curve of stability of this vessel is shown by A in Fig. 186, in which the range is seen to be 39°.

1. If now the beam of the vessel be increased by 7 ft., we should expect the GM to increase also, and this is seen by the curve of stability D starting from the origin at a steeper angle. This curve shows that, although the GM has increased from 2·6 ft. to 5·0 ft., owing to the increase of beam, yet the range of stability has only increased from 39° to 48°.

2. If instead of the beam we increase the freeboard by 7 ft., we obtain a strikingly different curve of stability, B. The increased freeboard is assumed not to affect the position of the C.G., and the

ship has therefore the same GM as at first, viz. 2·6 feet. The freeboard, however, has the effect of lengthening out the curve enormously, and the range is increased from 39° to 84°.

3. If now we take the ship as represented by C, assuming that the C.G. is raised 1 ft., making the GM 1·6 ft., the curve of stability is given by C, the range being reduced by about 10°. The curve C may be regarded as giving the double effect of increasing the freeboard, viz. rise of C.G. as well as increasing the freeboard.

We see from these examples—

(*a*) An increase of beam increases the initial stability, and

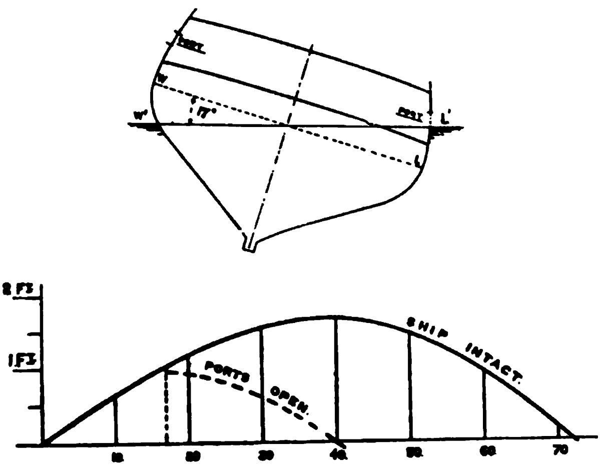


Fig. 187.—Stability curves, H.M.S. *Eurydice*.

therefore the slope of the curve near the origin, but does not greatly influence the area enclosed by the curve or the range.

(*b*) An increase of freeboard has a most important influence in lengthening out the curve and increasing the area enclosed.

(*c*) An alteration in the position of the C.G. influences both the initial stability and the area and range of the curve of stability. If the C.G. is raised to G′, say, then the ordinate of the curve at angle θ is lessened by GG′ sin θ. If the C.G. is depressed to G″, say, the ordinate of the curve at angle θ is increased by GG″ sin θ.

Dynamical Stability.—The work necessary to be done on a ship to force her over to a given angle is termed the dynamical stability at that angle. It can be shown that a measure of the

dynamical stability at any angle is obtained by the area of the curve of statical stability up to that angle. This is illustrated by Fig. 187, in which the curves of stability of the *Eurydice* are shown. When the sides of the ship were watertight to the upper deck, the curve of stability enclosed a large area. When however

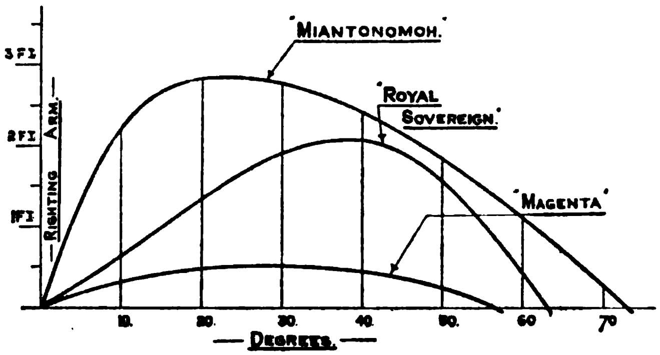


FIG. 188.—Curves of stability.

the ports were open, there was a sudden drop at 17°, and the ordinates of the curve decrease until the vanishing point was reached at 40°. The ratio of the areas enclosed is about 1 to 3; *i.e.* with the sides intact, about three times as much work would have

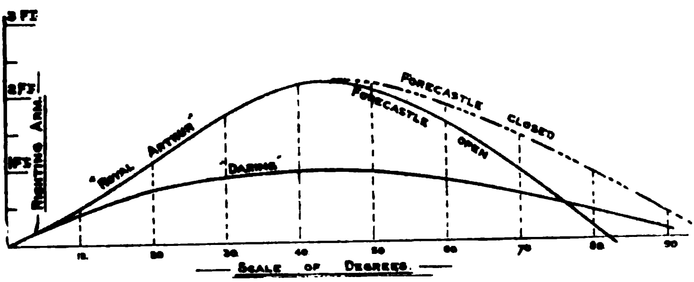


FIG. 189.—Curves of stability.

to be done by the wind and the heave of the sea to capsize the ship as would be necessary supposing the ports open.

It is thus seen that the area enclosed by a curve is of importance as well as the range of stability.

Curves of Stability.—In Figs. 188 and 189 are given some typical curves of stability.

The *Miantonomoh* was an American monitor of low free-board, viz. 3 ft., and great GM, viz. 14 ft. The curve rises with great steepness at the start, and the adverse influence of the low freeboard is counteracted by the great metacentric height.

The *Royal Sovereign* is a typical British battle-ship, with a GM of 3½ ft., and high freeboard, viz. 17 ft. The area enclosed by the curve is considerable, and the range is 63°. The *Triumph* and *Swiftsure*, recently bought into the Royal Navy, have a curve of stability very similar to *Royal Sovereign*.

The *Magenta* is a French battle-ship, with a GM of 2·3 ft., and high freeboard of about 16 ft. The low curve[1] for this vessel is caused by the small GM, and also by the great fall in of the side above water, which is a notable feature of ships of the French Navy. Fig. 132 shows a typical section of one of these ships, in

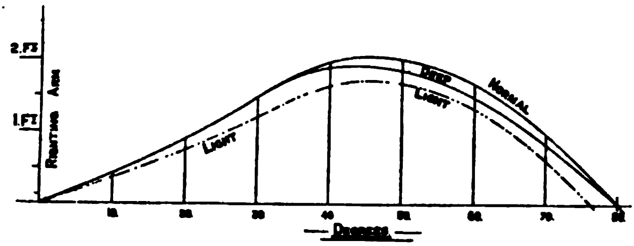


FIG. 190.—Stability curves.

which the breadth at the deck is only 48 ft., while the breadth at the waterline is 66½ ft.

The *Royal Arthur* is a typical British cruiser. For this ship two curves are constructed (*a*) assuming the forecastle open, and (*b*) assuming the forecastle closed. It is seen that without the forecastle the range is large, viz. 82°, and the effect of the fore-castle, if watertight, would be to increase the range to over 90°.

Destroyers have good curves of stability, as they have a relatively large GM and great reserve of buoyancy. The curve shown is for the *Daring*, in which the range is over 90°.

Any curve of stability of a ship is drawn for one particular displacement and position of the C.G., so that separate curves have to be drawn for the various assumed conditions of the ship, viz. deep load, normal load, and light, Fig. 190. The angles of maximum stability and vanishing stability in the two former

[1] *Marine Française*, May, 1895.

conditions are supplied to the Commanding Officer in the stability statement already mentioned. If the ship has a poop or forecastle, or both, the effect of these is usually noted on the statement. A specimen statement is given below. (It is suggested that the

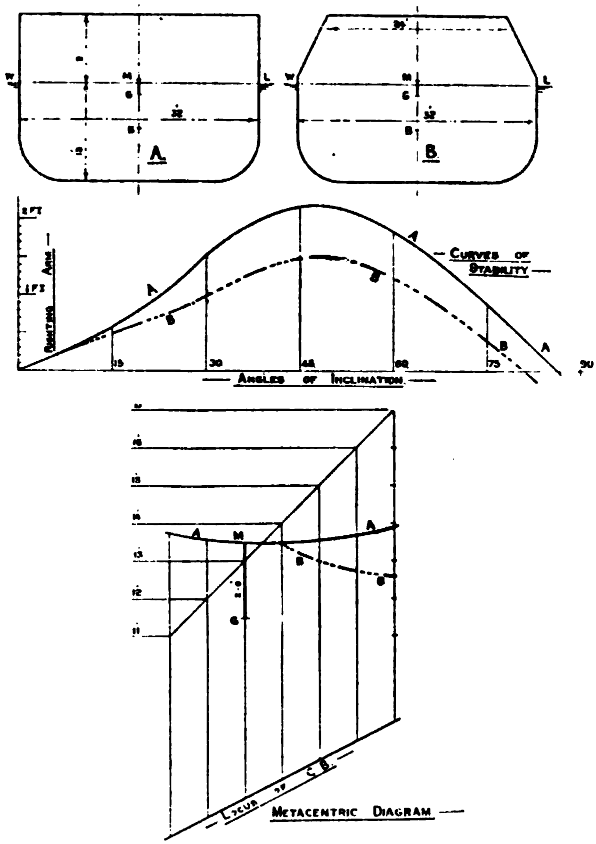


FIG. 191.

particulars of the stability of the ship in which officers are at present serving might be added.)

Effect of Tumble Home.—We have seen the importance of freeboard or reserve of buoyancy in giving ships a curve of stability with good range and enclosing a sufficient area. In the case, however, of some foreign ships, good freeboard is associated with a great tumble home, or fall in of the sides (see fig. 132).

This acts prejudicially in regard to the stability, as the following example will show:—

Two vessels have been taken, Fig. 191, both with 13 ft. draught, 9 ft. freeboard, 32 ft. beam, and 2 ft. metacentric height. In one case, however, the side falls in as shown to the deck, where the breadth is 24 ft. This results in a much lower curve of stability than is obtained without the fall in; we have seen that the area enclosed by a curve of stability is of great importance.

The corresponding metacentric diagrams are also interesting. In the second case, directly the side falls in the metacentric curve drops as the draught increases. Such a ship if damaged so that a bodily sinkage results might very possibly lose her initial stability, owing to this drop of the metacentre as the draught increases.

H.M.S. ————————.

STATEMENT OF METACENTRIC HEIGHTS AND STABILITY, BASED ON AN INCLINING EXPERIMENT MADE ON THE SHIP ON MARCH 27, 1900.

Conditions.	Feet.		Remarks.
A. The ship when fully equipped, with reserve feed-tanks empty, with 800 tons of coal in lower bunkers and 300 tons of coal in upper bunkers, at a mean draught of 20 ft. 6 in. has a metacentric height of ...	2·4		
B. The ship when fully equipped, with fresh-water and reserve feed-tanks and all bunkers quite full, *i.e.* with 1085 tons of coal on board, at a mean draught of 21 ft. 10 in. has a metacentric height of ...	2·4		
When lightened to a mean draught of 18 ft. 3 in., or when boilers are full to working height, engine condensers and feed-tanks at working height, and all coals, water (including reserve feed), provisions, officers' stores, and one half warrant officers' and engineers' stores consumed, the metacentric height is about	2·0		
	Forecastle.		
	Open.	Closed.	
	Deg.	Deg.	
The angle at which the ship reaches her maximum stability in the above condition A, and beyond which the righting force diminishes, is about	46	46	
The angle at which the ship reaches her maximum stability in the above condition B, and beyond which the righting force diminishes, is about	45	45	
The angle at which her stability entirely vanishes in the above condition A is about	80	93	
The angle at which her stability entirely vanishes in the above condition B is about	80	93	

NOTE.—*So far as stability is concerned the coal may be worked in any manner desired by the Commanding Officer.*

CHAPTER XX.

THE ROLLING OF SHIPS.

Rolling in Still Water.—Rolling in still water is of no immediate practical importance, because under ordinary circumstances a ship will not roll in still water. It is, however, necessary to study the subject, because it is only when the conditions operating in this case are understood that we are able to extend the inquiry to the more difficult case of rolling among waves.

If a ship, floating upright in stable equilibrium in still water, is inclined to a certain angle θ from the upright, the couple tending to take her back to the upright is $W \times GZ$. If the ship is released she will acquire angular velocity, passing through the upright to an angle on the other side rather less than θ. At this new angle, θ', say, she will have a couple tending to take her back to the upright, and so the ship, once being inclined and released, will continue to oscillate through smaller and smaller arcs of oscillation until she finally comes to a position of rest. When the ship is inclined to the angle θ as above, it is necessary to *do work* on the ship to effect the inclination, and this work is stored up in the ship as *potential energy*, or energy due to position. When the ship is released, this energy becomes converted into other forms of energy, and, if no resistances were operating tending to stop the motion, the ship, when passing through the upright, would have *kinetic energy*, or energy due to motion, exactly equal to the original potential energy.

This conversion of energy of one form into energy of other forms is well seen in the case of a stone on the top of a house. In some way work has had to be done to get the stone there, and the stone, in virtue of its position, has stored up in it a certain amount of potential energy. If the stone is released, it will reach the ground with a certain velocity, and (neglecting the friction of the air) the kinetic energy then possessed is equal to the original potential energy. When stopped by the ground both potential energy and kinetic energy disappear, but the energy is not lost, but is dissipated into the form of heat energy.

In the case of a ship rolling unresistedly the energy is alternately potential and kinetic at the extremity and middle of each roll, and the rolling would go on continuously; but when resistances operate, the energy gets drained away from the ship and becomes finally dissipated by imparting heat and motion to the air and water surrounding the ship.

Unresisted Rolling in Still Water.—It can be shown that for unresisted rolling in still water *the period of a single oscillation* (from port to starboard, or *vice versâ*) is very nearly given by

$$T = 0{\cdot}55\sqrt{\frac{k^2}{GM}} \text{ seconds}$$

where GM is the metacentric height and k is obtained from the following definition:—

The moment of inertia of a solid body about any axis is found by adding together the product of each weight making up the body and the *square* of its distance from the axis. (This is analogous to the moment of inertia of a plane area about an axis, dealt with in Chapter XVII.) If for a ship this axis is the axis of oscillation, W the weight and I the moment of inertia about the axis, then k is such a quantity that $I = W \times k^2$, and k is termed the *radius of gyration.*

The calculation for k is a most laborious one, but it has been done in a few cases, and having also the metacentric height, an estimate could then be made of the time of oscillation from the above formula. Practical agreement was found to exist between the actual and the estimated times of oscillation, even although the rolling could not have been unresisted.

The formula shows that to make the period *long*, *i.e.* to increase the time of oscillation, it is necessary to

(1) *increase* the radius of gyration, and, or
(2) *decrease* the metacentric height.

(The longer the period of a ship the more likely is she to be steady in a seaway.) Of these two, the first is of the lesser importance, because the distribution of the weights is governed by other features of the design than the desirability of obtaining a long period. The formula, however, shows clearly that *winging* weights, *i.e.* placing them at the sides, operates in the direction of increasing the period, although any practicable shift of weights on board a

war-ship can only have a small effect on the period. We should expect, therefore, to find that an armoured ship would roll more slowly than an unarmoured ship of about the same displacement and metacentric height, and this is confirmed by experience of ships in the Royal Navy.

A very considerable effect in lengthening the period is obtained by reducing the metacentric height. Thus in the *Royal Sovereign*, in which ship the period is 8 seconds and the GM about $3\frac{1}{2}$ ft., suppose the GM is reduced to 3 ft., without altering the radius of gyration. Then we should get a period of 8·64 seconds, or an increase of 8 per cent.

An interesting application of the above principles is found in the current practice of many merchant vessels. In many trades, voyages have to be undertaken with little or no cargo, because of the absence of return freights. It is necessary, for seaworthiness and proper immersion of the propellers, to sink the vessels by means of water-ballast. This has usually been placed in the double-bottom compartments. This, however, frequently pulls down the C.G. of the ship so far as to give the ship a large GM. This causes a very quick period, and in some cases this has not merely rendered the ship uncomfortable, but actually unsafe. In many ships, therefore, it is the practice to provide tanks in the 'tween decks and hold at the sides, and even on the upper deck. These tanks below are frequently large enough to hold ordinary cargo when necessary, but for "light" voyages they can be filled with water. The weight thus added, while giving sufficient immersion, does not produce excessive GM, and being at the sides tends to lengthen the period by increasing the radius of gyration.

The assumption used in obtaining the above formula for the period from side to side, viz. $T = 0{\cdot}55\sqrt{\frac{k^2}{GM}}$, is, that the righting lever varies directly as the angle or $GZ = GM \times \theta$, *i.e.* it assumes that the curve of stability is a straight line up to the angle considered. Under this condition *large and small inclinations will be performed in the same time.* A ship rolling in this manner is said to be *isochronous.*

Although the various assumptions made in obtaining the above formula are not strictly true, yet it is found by actual experiment that, within angles of 10° to 15° of the vertical, ships are very nearly *isochronous* in their rolling. This is the case although the ship experiences resistances which eventually bring her to rest.

The following are the approximate periods of some typical ships, *i.e.* the time from port to starboard, or *vice versâ.*

Inflexible	5¼ secs.	A vessel of large GM., viz. 8 ft., causing a small period. This was the cause of the introduction of water chambers (see later).
Royal Sovereign *Majestic*	8 secs.	Ships of moderate GM., about 3½ ft., and great moment of inertia, due to beam and armour at sides.
Powerful	7 secs.	Protected cruisers with no side armour.
Arrogant	6 secs.	
Pelorus	5½ secs.	
Gunboats and Destroyers	2–4 secs.	Small period due to (*a*) small moment of inertia; (*b*) relatively very large GM.

Resisted Rolling in Still Water.—Under the actual conditions under which a ship will roll in still water, resistances to the rolling are set up which drain the ship of energy and which sooner or later will bring her to rest. These resistances may be classified as follows:—

1. Friction of water on the ship's surface.
2. Effect of sharpness of form of ship's section.
3. Effect of bilge keels or keel projections (if any), including the flat portions of the ship.
4. Creation of waves on the surface.
5. Air resistance.
6. Use of water chambers.

1. **Friction.**—This cannot be of great amount in ordinary ships, because the surface is kept smoothly painted to reduce the resistance when steaming to the smallest possible amount.

2. **Form of Section.**—In a ship of circular section the relative velocity of the water and the surface of the ship is the same at all points of the section. In a ship of sharp form at the bilge, however, the water at the corner gets a motion opposite to the ship, and having to slip past the bilge, the effect both as regards friction and on bilge keels is greater in a sharp bilge than in a rounder form of section.

3. **Bilge Keels.**—The reason of the great extinctive effect of bilge keels in reducing rolling has been imperfectly understood until recently. The explanation is of considerable difficulty, and the following remarks do not pretend to completely deal with the subject:—

(*a*) A bilge keel is like a flat surface passing through water broadside on. The laws governing the resistance of such flat surfaces have been investigated, but in applying them to the case

of a ship it has been found that the extinctive effect observed could not thus be fully accounted for.

(*b*) A further influence has been suggested by Prof. Bryan, F.R.S. Consider the flow of water round a right angle as Fig. 192. The water next the surface has to suddenly change its direction at B. This causes a diminution of speed up to the point B, where it must be zero. The other streams of water are deflected, and along AB we get a diminution of velocity of the stream lines. This falling off of speed is accompanied by an increase of pressure, both along AB and BC.[1] If, for instance, a boat's rudder is suddenly put over to right angles, the leverage of the water pressure on the rudder about the axis of rotation of the boat is small, but the turning effect on the boat is considerable. This is caused by the pressure on the deadwood of the boat, which has a considerable leverage about the C.G. of the ship.

FIG. 192.

In the case of bilge keels projecting from the surface of a ship, suppose the ship is rotating clockwise, as in Fig. 193. The relative velocity of the ship and the water along A_2A_1 has to be brought to zero at A_1, and there is caused an increase of pressure along A_2A_1. This results in a resultant force P acting as shown, a similar force Q being found on the other side of the ship. These forces both act in such lines that they give a moment tending to stop the rotation. This effect will be more pronounced as the section of the ship is sharper, because of the greater relative velocity of the water past the bilge as compared with a round section.

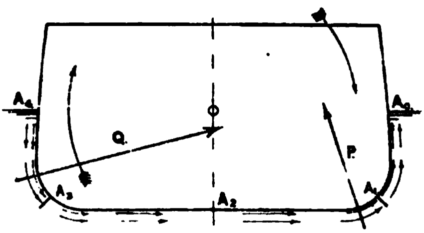


FIG. 193.

4. **Wave Formation.**—At each roll of the ship a wave is

[1] The relation between speed and pressure in flowing water, by which if speed diminishes the pressure increases, and *vice versâ*, is noticed in water-pipes. If a tap is turned off suddenly, as in the old-fashioned taps, a knock is heard in the pipes, caused by the sudden rise of pressure consequent on the motion being stopped. If, as in taps now in domestic use, the water is turned off gradually, the rise of pressure is not so sudden and the pipes are not so severely strained.

created on the surface of the water at each side; this wave passes away from the ship, and requires energy spent to create it. A wave of very small height represents a large amount of energy, and the drain on the ship's energy is a distinct resistance reducing the rolling.

5. Air Resistance.—The resistance of the air must be quite small under ordinary circumstances, but it may be made considerable by the use of steadying sails. It is well known that sails have a great steadying effect on a ship's rolling.

6. Water Chambers.—In the *Inflexible* and following ships a large metacentric height was an essential feature of the design, because it was necessary to provide such stability that the vessels should be able to stand upright, even supposing the unarmoured ends were completely riddled. It was known that this would cause a short period and quick rolling motion. This is an undesirable quality in any ship, and especially in a war-ship. The bilge keels could only have been of limited size because of the great beam of the ship causing difficulties in docking. On this account it was proposed and approved to fit athwartship chambers containing loose water. This water passes from side to side as the ship rolls, and causes waste of energy. This must be taken out of the ship, and so lessens the rolling. These water chambers were found to fulfil their purpose in diminishing the rolling, but the system was ultimately abandoned on account of the noise of the water rushing from side to side, and because the spaces were required for other purposes. As we have seen, there has been a gradual increase of waterplane area protected by armour in battle-ships since the *Inflexible*, so that the riddling of the ends has a less proportionate effect. On this account metacentric heights have been diminished from 8 ft. in that ship to $3\frac{1}{2}$ to 4 ft. in more recent ships. This has resulted in longer periods, so that the conditions of rolling are quite different, and the steadying effect of water chambers has not been required.

Bilge Keels.—Fig. 194 shows several forms of bilge keel as fitted to ships of the Navy. The standard form for steel ships is made of two $17\frac{1}{2}$-lb. ($\frac{7}{16}$ in.) plates connected to the ship's bottom as shown, the space between being filled in with light wood. The projection in battle-ships is inconvenient in connection with docking, and the breadth of the bilge keel is made somewhat less amidships than forward and aft where the ship gets narrower.

For smaller steel ships the bilge keel can be formed as shown; the figure gives the construction in a torpedo gunboat.

For sheathed ships the keel is formed of a single steel plate connected to the bottom by double angles, and supported at intervals by brackets. This is cased in with teak as shown.

All vessels in the Navy, including the latest destroyers, are fitted with bilge keels. The keels usually extend over rather less than half length.

Rolling among Waves.—In dealing with this subject it is important to note that a wave is not the passage of water but the passage of *motion*. The motion of the particles of water composing a wave is quite small, as may be noticed by watching a piece of wood among waves. The wave profile is seen to move along with considerable speed, but the wood sways backwards and for-

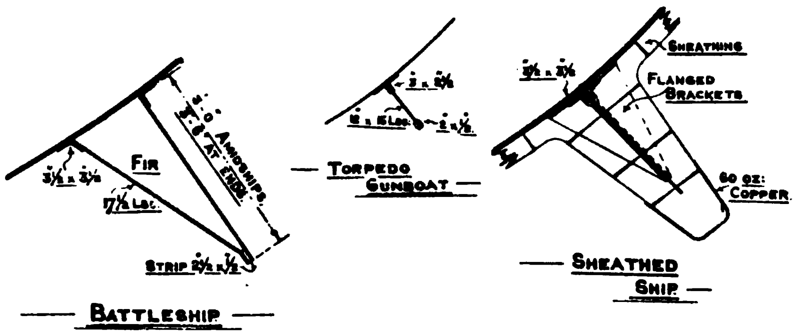


Fig. 194.—Bilge keels.

wards about a mean position. As a matter of fact the particles of water are moving in circular orbits, the radius of these orbits decreasing with the depth, so that at a moderate depth the water has a very slight motion. In waves, therefore, we find that the force of gravity is modified because of the centrifugal force set up by the orbital motion of the water.

It is well known that a can of water can be swung right round without any water spilling. When at the highest point the weight of the water acts down, but the circular motion gives rise to a centrifugal force acting outwards, and so long as this latter force is greater than the weight no water will be spilled. If the motion is slowed up a point would be reached when the weight would be greater than the centrifugal force, and the water would be spilled. It is the centrifugal force being greater than the force of gravity which keeps the car on the rails in "looping the loop."

At the crest of a wave the centrifugal force on the particles of water acts upwards *against* gravity, at the trough it acts downwards *with* gravity. The *apparent* or *virtual* weight of a body will therefore be less in the crest than the actual and more in the trough than the actual. This apparent weight may be 20 per cent. more or less than the real weight, according as it is in the trough or the crest.

This has been frequently verified by experience. If a spring balance is used, the indications on the dial for a given weight will be found less when the vessel is on the crest, and more when in the trough of a wave. This is also the explanation of the well-known phenomena of the tenderness of sailing boats on the crest of a long smooth wave. The virtual weight is considerably less than the actual, and consequently the righting moment is less than in still water. The wind moment is not affected in this way, and so on the crest of a wave, a boat, of sufficient stiffness in still water, is liable to be blown over to a large angle and possibly to capsize.

The effect of the centrifugal force at other portions of the wave is to alter the amount of the virtual gravity, and to cause it to

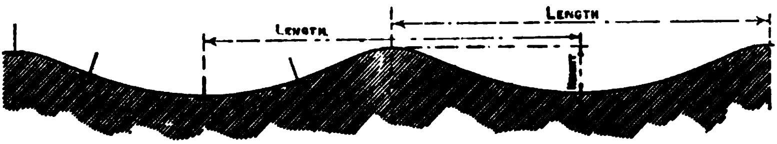


FIG. 195.—Wave profile.

act in a line of action perpendicular to the wave slope at any particular point. In considering rolling among waves we do not usually consider the variation of the virtual weight, but we must consider the variation of the line of action of the virtual buoyancy and gravity. This line of action will have its maximum inclination at about quarter the length of the wave from crest to crest or trough to trough. A small raft, as in Fig. 195, will always tend to keep normal to the wave surface; this normal is termed the *virtual upright* at any particular instant.

If now we take a ship floating broadside on to a series of waves (supposed long in comparison with the size of the ship), we shall have the set of forces as shown in Fig. 196. The inclination of the wave normal to the vertical is θ, and this wave normal is the virtual upright. If the ship is as shown the righting force is not due to the angle $\theta + \theta'$, but to the angle θ'.

If the ship has a very quick period compared with that of the wave, she will quickly come to the virtual upright, and so will take up the motion of the wave. This would be the case in a raft as in Fig. 195, and it is found that a ship of *very short period* does not roll very much or ship much water when rolling among waves, because she always keeps the deck parallel to the wave surface.

If a ship has a *long period* compared with that of the wave, the ship, at any particular instant, as in Fig. 196, does not come to the virtual upright with any suddenness, and the wave profile passes on and soon acts in the contrary direction. The ship therefore remains steady, never heeling to large angles. This quality of remaining nearly upright when among waves is termed *steadiness*, and is obtained in ships with a long period. We have seen above that a long period is mainly obtained by giving a small metacentric height.

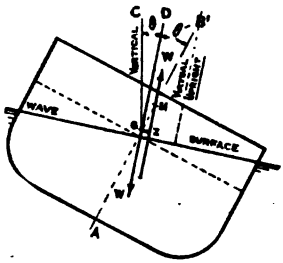


FIG. 196.

Such a ship is *crank*, *i.e.* is easily inclined by external forces, but in a seaway is most likely to be exceedingly steady.

If, however, a ship has her double period (port to port or *vice versâ*) equal to the period of the waves (time the length of the wave is traversed), we have a serious state of things. This timing is termed *synchronism*. As each wave passes the ship, an impulse is given timing with the period of the ship herself, and the tendency of this is to produce larger and larger angles of oscillation. If the vessel was perfectly isochronous for large as well as small angles, and no resistances were acting, such a system of waves would inevitably capsize her. The actual conditions operating, however, are as follows:—

1. As large angles are reached a ship departs from isochronous rolling,[1] and the condition of synchronism with the wave is not fulfilled.

2. Resistances operate, and, especially in a vessel with bilge keels, the energy imparted by the wave is soon absorbed by the energy taken out of the ship by the various resistances. When

[1] For a simple pendulum swinging 30° each side of the vertical 7 per cent. increase of period is noticed as compared with a small oscillation; for 45° the increase is 18 per cent.; for 60°, 37 per cent.

these are equal no further increment of rolling can take place.

3. A succession of waves of precisely the same period is a very unlikely occurrence.

It has often been noticed that ships with a great reputation for steadiness at sea occasionally roll heavily. This is doubtless caused by the fact that a succession of waves has been met with having a period approximately synchronizing with the double period of the ship.

In a ship thus rolling heavily a slight alteration of the course would be sufficient to destroy the synchronism, since what affects the ship is the *apparent* period of the waves, and if the ship's course be taken obliquely to the wave advance, the synchronism is at once destroyed.

The longer the period of a ship the less chance there is of meeting synchronizing waves. A series of waves of 16 seconds period is quite exceptional, so that the battle-ships of the British Navy having 16 seconds for their double period should be very steady, and this is borne out by actual experience. Atlantic storm waves have periods about 10 seconds, and it is only the smaller vessels of the Navy which have their double period as low as this; see above for periods of some typical ships.

Observations of Rolling.[1]—If a ship is rolling in still water, and a pendulum could be suspended at the centre of oscillation, then the point of suspension of such a pendulum would have no motion, and the pendulum would always remain vertical; the angles indicated would therefore give the angles of oscillation of the ship. If, however, the point of suspension is somewhere else, then as the ship rolls this point has motion and the pendulum does not give the true vertical. If one takes a fishing-rod, for instance, with a few feet of line, it is evident that, if the rod is swayed backwards and forwards, the line does not remain vertical. The same state of things obtains on board a ship; the pendulum does not hang vertically, and the angle it indicates will be in excess or defect of the true angle to which the ship rolls, unless it happens to be suspended at the centre of oscillation. If the point of suspension is above this centre the angle indicated will be in excess, if below, the angle will be in defect. (It is usually assumed that the centre of oscillation is near the C.G. of

[1] In Sir W. H. White's "Manual of Naval Architecture" a whole chapter is devoted to this subject.

the ship.) In the above illustration of the fishing-rod, if the line is very long, the motion of the rod does not have any sensible effect on the line, so that for all practical purposes the line will remain vertical. This is the principle which has been effectively employed in instruments for measuring rolling, viz. that a pendulum of *very long period* is not appreciably affected by the motion of the ship, but will maintain itself practically vertical as the ship rolls.

When a ship is rolling at sea, an ordinary pendulum is still less likely to give correct angles, as in addition to rolling, the ship has a bodily movement among the waves.

Mr. Froude's apparatus for rolling records is most valuable when accurate observations are desired, as it shows automatically the rolling of the ship together with a time record. It is, however, too elaborate an instrument for ordinary use. It primarily depends on a heavy wheel, so weighted that its C.G. is very close to the axis of suspension ($\frac{6}{1000}$ in. below), giving a very long period for a single oscillation,[1] viz. 34 seconds.

An instrument, Fig. 197, which is very simple and which has given admirable results has been devised by Mr. Mallock (I.N.A., 1901). It gives a pendulum of very long period, but the instrument is of small dimensions. A paddle P is supported on delicate pivots and is enclosed in a box containing fluid. The paddle is made of the same density as the fluid, and in this way the friction on the pivots is very small (buoyancy practically equals the weight). Any rotary motion of the outside case is not communicated to the mass of the fluid, and in the interior of the box the fluid is practically at rest when the box is in motion. The paddle is adjusted so that its C.G. is just below the

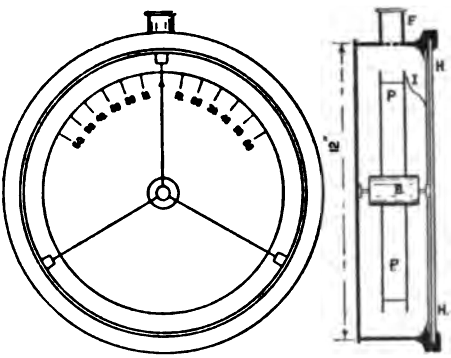


FIG. 197.—Mallock's rolling indicator.
HH, Glass front; B, Hollow boss; I, Pointer; F, Filling hole and expansion box.

[1] This is interesting as analogous to the case of a ship with great moment of inertia and small metacentric height, both of which conduce to a long period.

axis. When free it has a complete period of 4 seconds, but when enclosed in the box the complete period is between 30 and 40 seconds. The paddle therefore remains practically vertical, and the box being attached to the ship, the pointer I will show on the paddle the angle of roll.[1] This rolling indicator is now issued to ships of the Royal Navy.

The method of observing angles of roll by the use of battens is very simple. It can, however, only be used when the horizon or some fixed object (as a star) is visible. The battens are arranged so that they can be rigged up on the fore bridge as Fig. 198.

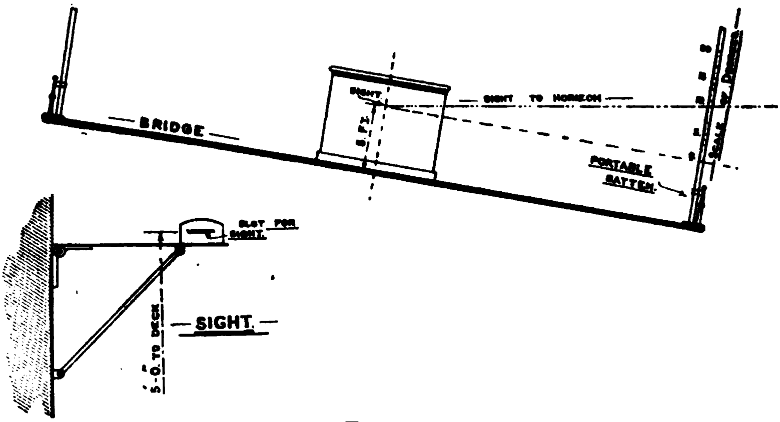


FIG. 198.

There are two battens on which a scale of angles is painted, the zero corresponding to the horizontal when the ship is upright. A bracket is placed at the middle line (attached to the chart house, say) having a horizontal slit. The horizon or distant object can be sighted through this slit at the extremity of each roll, and the angle can be noted. The time of each roll should also be noted by another observer. Forms are issued to ships of the Royal Navy giving detailed instructions.

[1] "Mr. Mallock's instrument is exceedingly simple, it is always in place, it may be put anywhere, it is always measuring the angle of heel and is ready to be observed. I have invariably heard it spoken of very highly by those who have used it."—Mr. Philip Watts, F.R.S. (I.N.A., 1901).

CHAPTER XXI.

THE TURNING OF SHIPS.

WHEN the rudder of a ship moving ahead is put over, a force is brought into existence causing the ship to (1) heel, (2) turn, (3) to slacken in speed, and (4) to have side movement or drift. The rudder, being placed obliquely to the middle line of the ship, causes the streams of water flowing aft to be deflected, and this causes a force to act upon the rudder, as P, Fig. 199. The value of this normal force depends upon the area of the rudder, the *square* of the speed of the water meeting the rudder, and the angle to which the rudder is placed. In a sailing-ship the speed of water meeting the rudder is rather less than the speed of the ship, because the friction of the ship's surface causes a layer of water to be dragged along in the direction of the ship's motion. The rudder of such a ship is thus not passing through still water but through water which has a forward motion. The steering of a sailing-ship depends on the motion of the ship, and such a ship loses her power of steering as she loses way. With a screw-steamer, although there is the same *frictional wake*, yet the action of the propellers send a stream of water astern, and such a ship has steerage directly the engines are working, before she attains any motion at all. A ship with a very full stern is likely to steer badly, as the water does not get a clean flow past the rudder, which is necessary in order to get the normal pressure required. (See discussion of the steering of *Agamemnon*, United Service Institution, 1887–88.)

In Fig. 199, let P be the normal pressure acting on the rudder at C. At the C.G. of the ship, G, introduce two equal and opposite forces, P, in a line parallel to the line of action of P. Then we have acting on the ship—

(i.) A couple tending to turn the ship, as shown, of magnitude $P \times DG$; and

(ii.) A force, P, acting in the line EG.

This force, P, will have a transverse component, FG, P cos θ, tending to move the ship bodily to starboard, and a fore-and-aft component, EF, P sin θ, tending to stop the ship. The force causing side motion has small effect, since the resistance of the ship to this motion is very great. The fore-and-aft component has, however, a sensible effect in checking the speed when turning.

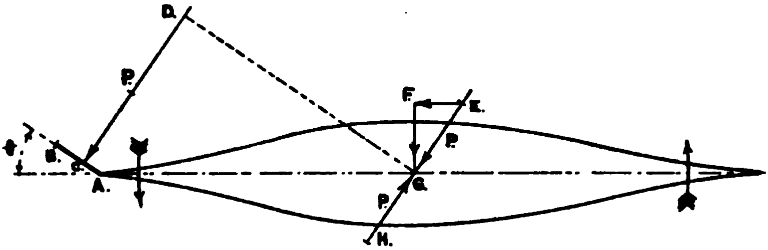


FIG. 199.

In a ship with a deadwood there is a side pressure due to the slackening of the stream lines on putting the rudder over. This side pressure on the deadwood has a considerable leverage to turn the ship (see Fig. 192, and note on action of bilge keels, p. 226.)

Heel caused by putting Rudder over.—On first putting a rudder over, the force P has a tendency to cause heeling inwards. This inward heeling is specially felt in vessels like destroyers, in which the rudder area is relatively large. In a full-sized ship, however, this inward heeling tendency is only of short duration, as when the ship gets on the circle the centrifugal force comes into action, and when, as is usually the case, the C.G. of the ship is above the centre of pressure of the water on the outward side (centre of lateral resistance), there is a couple, as shown in Fig. 200, tending to heel the ship outwards. This heeling tendency is resisted by the stability of the ship, and it can be shown that the vessel will take up an angle of heel θ, given approximately by—

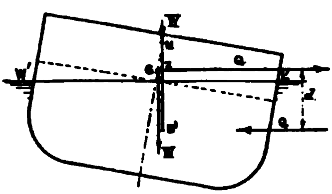


FIG. 200.

$$\sin \theta = \frac{1}{11}\left(\frac{d}{GM} \times \frac{V^2}{R}\right)$$

where d is the distance in feet between the C.G. of ship and the centre of lateral resistance;

V is speed in knots on the circle;

R is radius of turning circle in feet;

GM is the metacentric height.

The above shows the qualities of a ship which affect the heeling when on the circle, viz.—

(i.) It depends *directly* as the *square* of the speed;

(ii.) It depends *inversely* as the radius of the turning circle; and

(iii.) It depends *inversely* as the metacentric height.

Thus a ship of high speed and small GM, turning in a small circle, might possibly heel to a considerable angle, sufficient to prevent the guns being laid horizontal on the inner side.

In the case of *Yashima*,[1] a Japanese battle-ship, the outward heel at full speed was $8\frac{3}{4}°$, at 10 knots only 2°. This ship had a very large rudder, and turned in a very small circle. On first putting the rudder over there was an inward heel, but when on the circle the inward heel, due to the pressure on the rudder, was overcome by the outward heel, due to the centrifugal force.

In destroyers, where the distance of the C.G. from the centre of lateral resistance is not great, the outward heeling tendency due to centrifugal force on the circle may be overcome by the inward heeling due to the rudder pressure. If the helm in such a ship were suddenly righted, the inward heeling tendency, due to the rudder, would be suddenly withdrawn, and the ship might give a dangerous lurch outwards. Under these circumstances the rudder should be righted gradually, so that as the rudder pressure is withdrawn the ship may come off the circular path. The above is one of the reasons for giving destroyers a relatively large metacentric height, in order to provide for their safety when manœuvring.

Pivoting Point and Drift Angle.—In a ship turning in a circular arc the centre line of the ship points inside the circle, so that the thrust of the propellers is delivered in a direction oblique to the motion of the ship. This, together with the drag of the rudder, is the reason of the reduction of speed always experienced when turning. If, at any instant, the ship is as shown in Fig. 201, G_1GG_2 being the path of the C.G. and O being the centre of the path, then GT being the tangent to the path at G, the angle PGT is the *drift angle* at the point G. At the point P, where OP is drawn perpendicular to the centre line of the ship, there is no

[1] See a paper by Mr. Philip Watts, F.R.S. (I.N.A., 1898).

drift angle, as the tangent to the circle through P is the centre line of the ship.

The motion of any point in the ship is instantaneously in the direction of the tangent to the circle that point is turning in. At the point P this tangent is the centre line of the ship. At the point *b*, for instance, the motion in the direction *bc* may be resolved into its components, *bd* in the direction of the keel, and *be* in an athwartship direction. All points abaft P therefore will, *relative to* P, move to port, and all points forward of P will move to starboard in Fig. 201. This is why, to an observer on board, the ship appears to be turning about the pivoting point P.

Path of Ship when turning.—When the rudder of a ship is

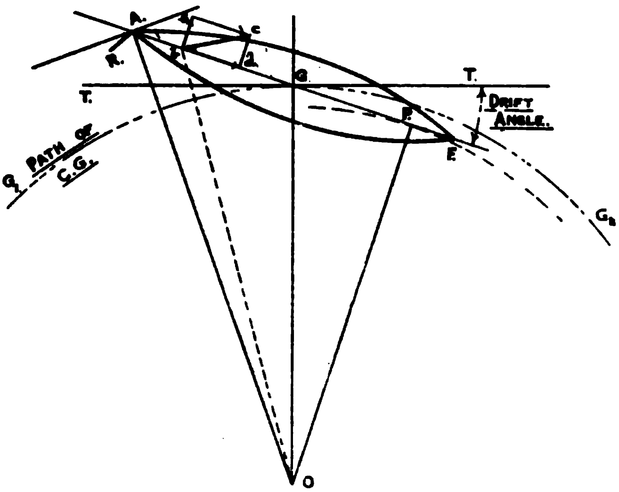


FIG. 201.

put over, the ship commences to turn in a spiral path, as Fig. 202. By the time she has gone through eight points the path is approximately circular. The distance from the position at which the helm is put over to the position when she is at right angles to her original course is termed the *advance*, and the distance from the original course to the position of the ship when she has turned through sixteen points is termed the *tactical diameter*. The path swept out by the stern will have a greater diameter than this. This must be allowed for when considering the room a ship can turn in.

The features of a ship which influence the turning are—

1. Time of putting the helm over.
2. Angle of helm.
3. Size of rudder.
4. Moment of resistance of underwater body of ship to turning.
5. Moment of inertia of the vessel.

1. In modern ships with steam steering gear the time of putting the rudder hard over is a matter of seconds only. The shape of the rudder is of importance in this connection; a rudder that is balanced has the centre of pressure close to the axis, and offers a small resistance only to being put over. Such a rudder can be got over more quickly than one hinged on the fore side (see rudders in Figs. 71 to 76). The more quickly the rudder is put over the sooner its turning effect comes into operation. This is illustrated by the paths on turning of *Orlando* and *Astræa* (Fig. 202). Although the former ship has a smaller tactical diameter, yet it is longer in getting into the circle because of the fact that the rudder was not balanced. If there is a difference in the time of getting the rudder over a thigh speeds as compared with low speeds, the tactical diameter at the higher speeds will be greater than at the lower speeds. Usually, however, with steam steering gear, the path on turning is practically constant for all speeds.

2. The usual maximum angle of helm in ships of the Royal Navy is 35°. The tactical diameter will vary approximately *inversely* as the angle of helm, so that a vessel may be made to turn in a path greater than that with the maximum helm angle by using a smaller helm angle. Ships of different type may thus be made to move through similar arcs by determining the helm angles beforehand by experiment.

3. The size of the rudder has a direct influence on turning, because the pressure P depends directly on the rudder area. This area is expressed as a fraction of the area of the immersed middle line plane of the ship. For large ships in the Navy this ratio is from $\frac{1}{40}$ to $\frac{1}{50}$. Recent battle-ships and cruisers have a ratio of $\frac{1}{45}$, about. The *Yashima* mentioned above has the ratio $\frac{1}{35}$. In a typical destroyer the ratio was $\frac{1}{33}$.

4. The resistance the ship offers to turning depends on the shape of the underwater body and the position of the "pivoting point." This pivoting point is usually forward of amidships when going ahead; in some ships it is right up at the bow. The

moment of resistance offered by a portion of the surface will vary roughly as the *cube* of its distance from the pivoting point, and as

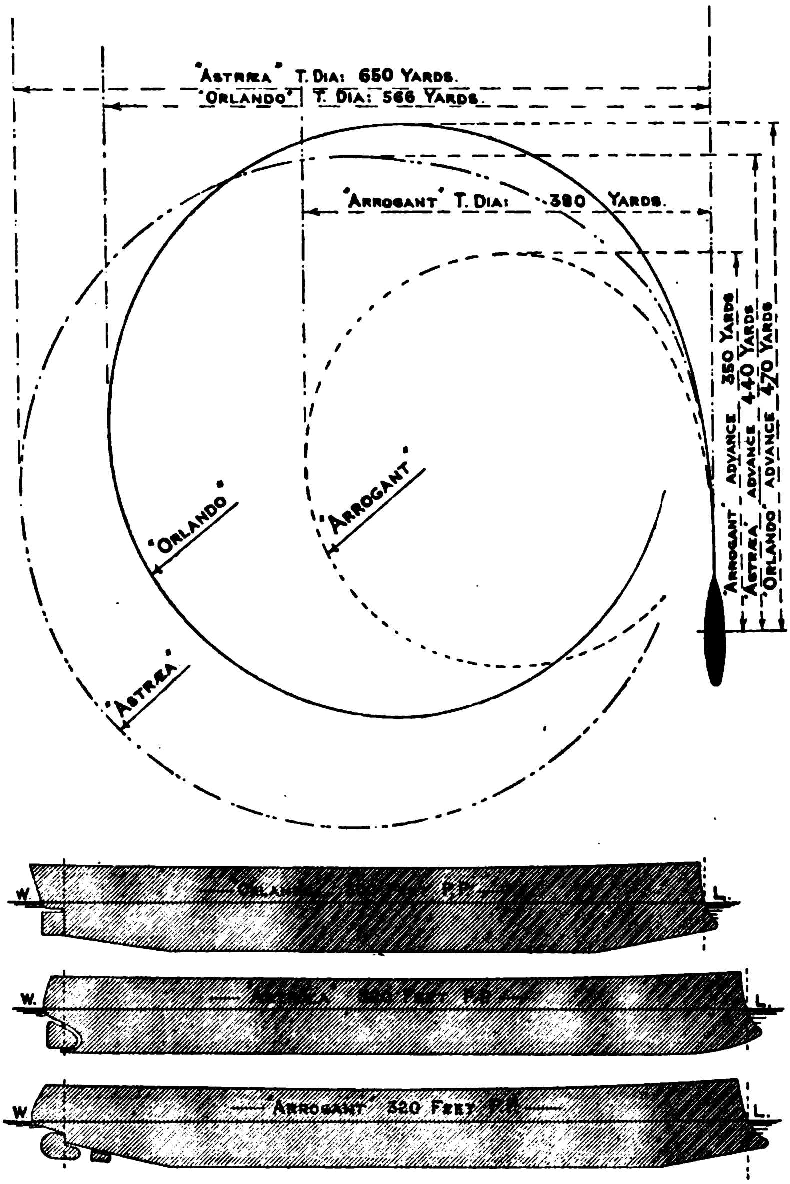


Fig. 202.

the cosine of the angle it makes with the vertical. The flat portions of a ship at the after end are therefore best adapted for offering effective resistance to turning, and on this account the flat portions at the stern of recent ships are cut away in order to improve the turning (see Figs. 72 to 76 for examples). In large cruisers the stern is cut right up, with an underhung balanced rudder. In the fourteen battle-ships of *Formidable* and *Duncan* classes the stern is cut away as shown in Fig. 75, being brought down at the sternpost to take the blocks when docking. In *King Edward VII.* (Fig. 76) this cut away is associated with a partially balanced rudder. The cut up at the bow is of little value in influencing the turning when going ahead, but has some influence when going astern. A ship trimming by the stern more than usual will have a larger tactical diameter in that condition, and the converse will be the case if she trims more by the head. A short ship will turn more readily than a long ship, on account of the less resistance offered to the turning.

5. A ship with heavy weights at the extremities will turn more slowly than a ship of the same size and weight, etc., with the weight concentrated more amidships, and when once turning will be more difficult to get back to the straight again. This is due to the greater moment of inertia of the ship about a vertical axis in the former case.

Suppose two balls each weighing 1 lb. are fastened on a stick 12 in. apart, and two other balls of the same weight are fastened on a stick 60 in. apart. It is readily seen that the latter system is more difficult to start rotating about an axis in the middle perpendicular to the stick than the former, and when once in motion will be more difficult to stop. This is due to the different *moments of inertia.* In the first case it is roughly $2[1\times(\frac{1}{2})^2]$ (*i.e.* weight multiplied by square of distance), in the second it is roughly $2[1\times(\frac{5}{2})^2]$, the ratio being $(\frac{1}{2}) : (\frac{25}{2})$, or 1 : 25.

The comparison between the turning circles of *Orlando*, *Astræa*, and *Arrogant* (Fig. 202) illustrates the above principles very clearly. The *Orlando* is 300 ft. long, with a rectangular rudder. The *Astræa* is 320 ft. long, with a balanced rudder and no cut up at the stern. The *Arrogant* is also 320 ft. long, but has two balanced rudders with considerable cut up at the stern.

The *Orlando*, although shorter than the *Astræa*, does not get into the circular path so soon as the *Astræa*, on account of the type of rudder, which is not balanced. She turns, however, in a smaller circle than the *Astræa* on account of the lesser length.

The difference between the *Astræa* and the *Arrogant*, both of the same length, is very marked.

The *advance* of *Arrogant* is 350 yards, of *Astræa*, 440 yards.

The *tactical diameter* of *Arrogant* is 380 yards, or 3·6 lengths.

" " " *Astræa* is 650 yards, or 6·1 lengths.

This great difference is due to two causes, viz. the double rudders of *Arrogant* and the large cut up at the stern of that ship. The ships of the *Arrogant* class were specially designed as fleet cruisers, and this great turning facility was made a feature of the design.

The following comparison between the turning of the *Diadem* and *Cressy* illustrates also the influence of the shape of the stern. The *Diadem* is 435 ft. × 69 ft. × 25¼ ft. × 11,000 tons, with a stern like *Edgar* in Fig. 71. The *Cressy* is 440 ft. × 69½ ft. × 26¼ ft. × 12,000 tons, with a stern shaped as shown in Fig. 73. The rudder of the *Cressy* is rather larger than in *Diadem*, but the ratio of rudder area to immersed middle line plane is the same in both cases.

	Tactical diameter.		Time to 8 points (seconds).
	Yards.	In terms of ship's length.	
Diadem	914	6·3	101
Cressy	671	4·6	74

Turning of a Twin Screw Ship.—The above discussion deals with the turning of ships under the action of the rudder alone. A twin screw vessel, however, may be made to turn in a smaller arc by the use of its screws in association with the rudder. The engine on the side to which the rudder is put would be worked ahead, and the other worked astern. This power of turning in the smallest possible circle may be of great value in special circumstances to avoid collision. It is found that the advance with one screw ahead and one astern is about 70 to 80 per cent. of the advance with both screws ahead. The tactical diameter is about 60 to 70 per cent.

Twin screw vessels have a great advantage over single screw ships because of the possibility of steering by the screws alone, by varying the revolutions. Several battle-ships have recently gone long voyages without a rudder at all, the steering being done by the twin screws.

Turning Trials.—Systematic turning trials are carried out on all H.M. ships, and a record is kept in the ship's book for the information of those officers who have subsequently to navigate the ship. There are two sets of trials; the first those carried out during the official steam trials of the ship when she is in the dockyard reserve, and secondly a series of turning trials at 12 knots and 6 knots, carried out when the ship is in commission.

1. *Trials in dockyard reserve.*—Most of these are to determine the time of turning, the advance and tactical diameter at the full natural draught power; but some of the trials determine these also for the speed of 10 knots.

2. *Trials when in commission.*[1]—The trials ordered to be carried out are divided into four sections.

(1) At 12 knots with full helm.
At 12 knots with full helm and one engine at the revolutions for 12 knots astern.
(2) At 12 knots with 25° and 15° of helm.
(3) At 6 knots with full helm.
(4) With helm amidships to determine the time and distance before the ship loses way.
(*a*) With engines at 12 knots and then stopped.
(*b*) With engines at 12 knots and then reversed with all steam at command.
(*c*) With engines at 6 knots and then reversed with all steam at command.

The object of (*b*) and (*c*) of the last section is to ascertain whether the ship can best avoid an object right ahead (as shallow water or another ship) by reversing with all steam at command, or by turning with both screws ahead, or with one screw reversed as in section (1).

It is laid down that each section should be completed in a day, and if possible all the four sections should be undertaken with the ship in similar conditions of trim, in similar weather, and in water over 20 fathoms deep. Full instructions how to proceed with the trials are contained in the form No. S. 347.

In using a range-finder for getting the distances of the buoy from the ship, notice must be taken of the lower limit of the range-finder, so as not to go too close to the buoy.

[1] Previous to 1902 the trials were somewhat different; the principal change has been in the alteration of speed, then 10 and 5 knots, now 12 and 6 knots.

CHAPTER XXII.

THE RESISTANCE AND PROPULSION OF SHIPS.

Resistance.—The resistance opposed to a ship when moving through water is much more complex than the resistance offered to the motion of a train, say. In first considering the subject, we must leave out of account the disturbance caused by the propelling agent, usually the screw propeller, and imagine that the ship is towed through the water by some other ship. This has actually been done by experimenters on the subject, the most notable series of experiments being those carried out by Mr. W. Froude on H.M.S. *Greyhound* in 1871. Mr. Froude had the ship towed by H.M.S. *Active*, as in Fig. 203, to avoid any disturbance due to the wake behind the latter ship. The tow-rope was connected on the

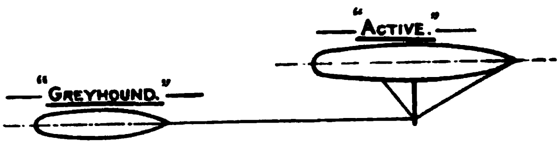


FIG. 203.

Greyhound to a dynamometer, to register the strain, and it was this strain which was overcoming instant by instant the resistance offered by the water to the onward motion of the *Greyhound*. The experiments were carried out over a wide range of speed, and as a result Mr. Froude had a record of resistances at various speeds. When such a record as this is obtained, it is convenient to represent it graphically by drawing a base to represent speeds, and erect ordinates to represent the resistances. The spots thus obtained enable a curve to be drawn showing resistance on a speed base. The curve obtained for the *Greyhound* is shown by AA in Fig. 204, and it is very suggestive. We notice that the

resistance does not increase regularly as the speed increases, but the *rate* of increase is much more rapid at high speeds than at low speeds. Thus to increase the speed from 7 to 8 knots an extra resistance of 1500 lbs. has to be overcome, whereas to increase the speed from 11 to 12 knots an extra resistance of 6000 lbs. has to be overcome, or four times as much for an increment of 1 knot.

This agrees with our experience. We know how much more difficult it is to increase the speed of a ship by a knot, say, near the top speed than at the lower speeds. Fig. 205 shows the curve of I.H.P. on base of speed of H.M.S. *Drake*, and the following

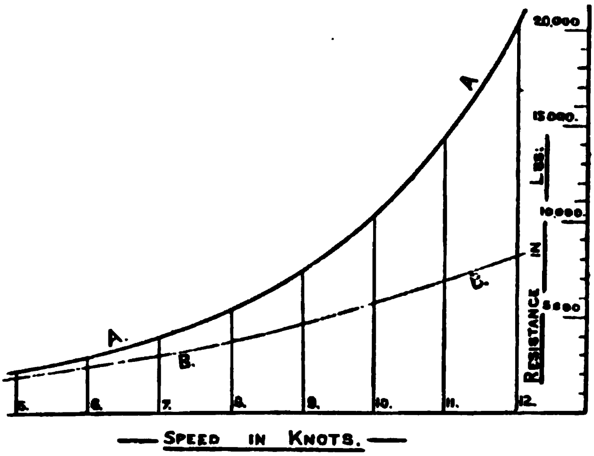


FIG. 204.

shows in tabular form the increase of power necessary for each 2 knots from 10 to 24 knots:—

Speed in knots . .	10	12	14	16	18	20	22	24
I.H.P.	1,950	3,200	4,800	7,000	10,000	14,800	21,900	31,000
Additional I.H.P. necessary for each increment of 2 knots		1,250	1,600	2,200	3,000	4,800	7,100	9,100

To increase the speed from 22 to 24 knots requires as much power as is sufficient to drive the ship $17\frac{1}{2}$ knots, and to increase the speed from 20 to 24 knots means more than doubling the

horse-power. This great increase of power necessary for high speeds is due to the great increase of resistance.

The *rate* at which resistance increases as speed increases is therefore a matter of great importance. Mr. Froude found in the *Greyhound* that up to 8 knots the resistance was varying as the

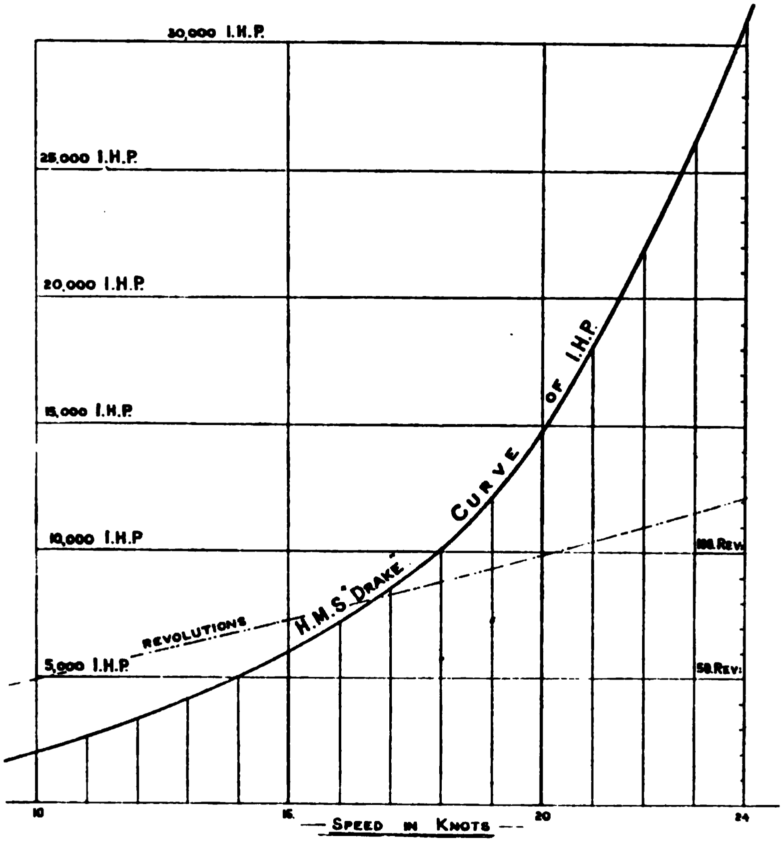


FIG. 205.

square of the speed. That is to say, if R_1 is the resistance at speed V_1 and R_2 is the resistance at speed V_2, then—

$$R_1 : R_2 :: (V_1)^2 : (V_2)^2$$

$$\text{or } \frac{R_1}{R_2} = \left(\frac{V_1}{V_2}\right)^2$$

Mr. Froude found, as the curve in Fig. 204 indicates, that the

resistance varied much faster than the square of the speed at the higher speeds. Indeed, at 12 knots it was varying as the *fourth* power of the speed, a very high rate of increase. In the case of the *Drake* the resistance is varying nearly as the third power of the speed between 23 and 24 knots.

Effective Horse-power.—If we know the resistance of a ship which is being towed at any given speed, we can determine the horse-power[1] that is being transmitted through the tow-rope to overcome this resistance. This is the effective horse-power, sometimes called the tow-rope horse-power. This horse-power is a very different thing from the power exerted by the vessel's own engines or the *indicated horse-power* (I.H.P.).

In any general case, if R is the resistance in pounds, V the speed in knots (1 knot is a speed of 6080 feet per hour), then—

$$\text{Work done per minute} = R \times \left(\frac{V \times 6080}{60}\right) \text{foot-lbs.}$$

$$\text{and horse-power} = \frac{R \times \left(\frac{V \times 6080}{60}\right)}{33{,}000}$$
$$= \tfrac{1}{326}(R \times V).$$

This is the effective horse-power (E.H.P.). Mr. Froude was thus able at once to turn the resistance of the *Greyhound* at any speed into E.H.P. He found a striking difference between the E.H.P. thus obtained and the I.H.P. which had to be exerted by the vessel's own engines in order to get similar speeds. Thus at 10 knots the E.H.P. worked out to 380, and for this speed the I.H.P. necessary was 786, giving a ratio of E.H.P. ÷ I.H.P. of $\frac{380}{786} = 0{\cdot}42$. This was a most important result, showing that of the power exerted at the vessel's own engines the large proportion of 58 per cent. was wasted so far as the ship was concerned. Mr. Froude, on arriving at these results, was led to make further investigations in order to look into the cause of this great loss of power.

The usual value of this ratio in vessels of the Royal Navy is from 45 to 50 per cent., rising to 55 per cent., or higher in some

[1] The *work* done by a force (like the strain in the tow-rope), acting through a certain distance, is given by the product of the force and the distance through which it acts. This is independent of the *time* taken. The *power* exerted takes into account the *time* in which the work is done. The unit of power used is the *horse-power*, which is defined as 33,000 foot lbs. of work performed in one minute.

few cases. This ratio is termed the *propulsive coefficient*, being the ratio of the horse-power usefully employed to the horse-power actually exerted.

We have been considering above the total resistance experienced by a ship on being towed through water, it is necessary now to inquire how this total resistance is made up. It may be divided into four parts, viz.—

1. Resistance due to the friction of water on the surface.
2. Resistance due to the formation of eddies.
3. Resistance due to the formation of waves on the surface.
4. Resistance of the air.

1. Frictional Resistance.—This can be directly calculated from data obtained from a series of experiments made by Mr. Froude on boards coated with various surfaces. For an ordinary smooth surface like a vessel coated with paint the resistance in pounds is given by—

$$R = f . S . V^{1\cdot83}$$

where S is wetted surface in square feet;
V is speed in knots;
f is a coefficient.

This coefficient varies according to the length of the surface, being greater for short than for long surfaces. For ships its value does not vary much from 0·009. For short models such as are used in the experimental tank the value of f is greater. We notice that this resistance for a smooth surface varies at a rather lower rate than the square, viz. 1·83. If, however, the surface is rough, like sand, the coefficient f is twice as great, and the power of the speed rises from 1·83 to 2·0. This illustrates the fact that in order to keep the resistance as low as possible, and so economize horse-power, and therefore coals, it is necessary to keep the bottom clean by periodical docking. On this account, also, ships which are employed on distant service, with the probability of remaining at sea for long periods, are sheathed with copper to prevent fouling.

The frictional resistance is of importance at all speeds, but at low speeds it accounts for the bulk of the resistance. For a torpedo-boat destroyer, which has an abnormally wide range of speed, at 12 knots the frictional resistance is 80 per cent. of the total; at 16 knots, 70 per cent.; at 20 knots, 50 per cent.; and at 30 knots, 45 per cent.

2. Eddy Resistance.—This is due to the eddies formed behind

a blunt ending to the underwater body. Ships built as formerly, with very full sterns and thick sternposts, experienced this resistance to a large extent, but in modern ships of finer form it does not exist to any appreciable degree. Every care is taken to avoid any abrupt terminations which might cause eddy making. One instance of this is seen in the shaft brackets; the section of the arms is made as in Fig. 82, taken to a small radius at the after end, so that no eddies are caused at the rear.

3. **Wave Resistance.**—It is this form of resistance which becomes of the greatest importance at high speeds, and it is because of the rapid growth of this resistance that it becomes increasingly difficult to obtain these high speeds in full-sized ships.

When a ship is towed through water there are two separate and distinct series of waves brought into existence, viz. those formed at the bow and those formed at the stern. Each of these series consists of (*a*) a series that diverge with their crests sloping aft, and (*b*) a series of transverse waves whose crests are nearly perpendicular to the middle line of the ship.

The diverging waves both at the bow and the stern at once pass away from the ship. The transverse waves of the bow series are of the most importance, and the interference of these waves with the corresponding waves at the stern causes considerable variation of the resistance. If a crest of the bow wave series coincides with a crest of the stern wave series there is an increase of wave resistance. A decrease is found to result if a crest of the bow wave series coincides with a trough of the stern series.

When a ship maintains a steady speed, say 15 knots, the accompanying series of transverse waves also has a speed of 15 knots. Such a series has a definite length, viz. 126 ft. from crest to crest, or trough to trough. At 10 knots the length would be 56 ft. It is thus possible to make an estimate of the speed at which a ship is travelling by observing the length from crest to crest of the wave along the side of the vessel. (If V be the speed in knots and L the length of wave in feet, then $V = 1{\cdot}33\sqrt{L}$.) As speeds therefore increase, the accompanying wave system gets longer, increasing in length as the *square of the speed*. In the case of small vessels, like destroyers, travelling at the high speed of 30 knots, say, a wave is created longer than the ship, and she lies on the back slope of a wave of her own creation.

We thus see that as speeds increase, the length and height of the waves formed must also increase very rapidly, and consequently

the energy required to maintain them. The resistance thus caused varies at a higher rate of the speed than the square, and it is not possible to determine beforehand by calculation its amount for any given case. We have to rely on the results of model experiments, or on the trials of ships of similar form, when estimating the power necessary for a new design of high speed.

4. Air Resistance. — This is a subject about which our knowledge is very scanty. It became of less importance than formerly with the abolition of sails, but in the present high-speed ships it must be of appreciable amount. If a ship is steaming against the wind, the relative velocity is the speed of wind plus speed of ship, and this gives rise to considerable resistance.

Corresponding Speeds.—We continually use the terms high speed and low speed, as applied to certain ships, but these terms are strictly relative. What would be a high speed for one ship might very well be a low speed for another. Thus 15 knots is a high speed for a ship 150 ft. long, but quite moderate for a ship 500 ft. long. To obtain a real measure of speed in any case we find its ratio to the square root of the length, viz. $\frac{V}{\sqrt{L}}$

where $\frac{V}{\sqrt{L}} = 0{\cdot}5$ to $0{\cdot}7$ the ship is being driven at a moderate economical speed;

$\frac{V}{\sqrt{L}} = 0{\cdot}7$ to $1{\cdot}0$ we have the speed of mail steamers and battle-ships;

$\frac{V}{\sqrt{L}} = 1{\cdot}0$ to $1{\cdot}3$ we have cruiser speeds.

Beyond this we cannot go in full-sized vessels under present conditions, because it is not possible to get enough engine and boiler power into the ships. It can be done, however, in destroyers by using very fast-running engines and forced draught to boilers, and in these ships $\frac{V}{\sqrt{L}} = 1{\cdot}9$ to $2{\cdot}5$. Such speeds as these are excessive, and require a great expenditure of horse-power to obtain.

"When these excessive speeds are reached, although the horse-power required is very great, yet the resistance does not vary at so great a rate as is the case at lower speeds. The following figures show how the total resistance varies in a typical destroyer:—

Up to 11 knots	it varies	as the	2nd	power	nearly.
At 16	"	"	3rd	"	"
From 18–20	"	"	3·3rd	"	"
At 22	"	"	2·7th	"	"
At 25	"	"	2nd	"	"
At 30	"	"	2nd	"	"

If we took the destroyer as a model, and took a vessel of the same form but 14,100 tons displacement, 25 knots of the destroyer would *correspond* to 47½ knots of the larger vessel, and this vessel would not reach the condition where further increments of speed are obtained with comparatively moderate additions of power until she reached 47 knots, which is an impossible speed for such a vessel under existing conditions" (Sir W. H. White's Address to British Association, 1899).

As speeds have continually increased in the mercantile marine, the lengths have also shown a corresponding increase, one result of which is to keep the ratio $\frac{V}{\sqrt{L}}$ within moderate limits. In cruisers, however, this continual increase of length has been undesirable as speeds have increased and the ships are only of

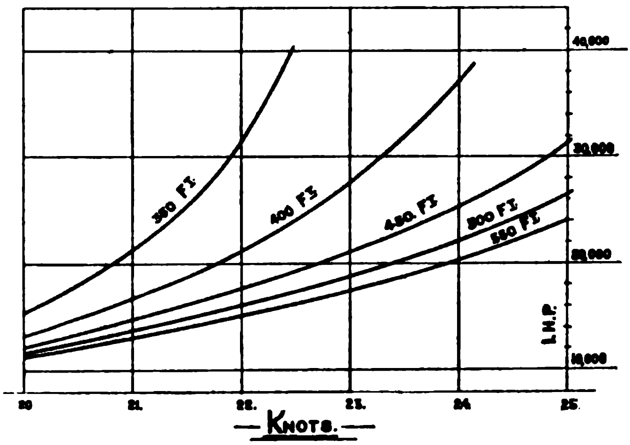


FIG. 206.
I.H.P. of ships of 9000 tons displacement, but of varying lengths.
(*From Mr. Watts's article in "Encyclopædia Britannica."*)

sufficient length to obtain the qualities desired, and the extra power rendered necessary by limiting the length has had to be accepted. If lengths had been increased the economy of propulsion obtained would have been discounted by losses and disadvantages in other directions.[1]

[1] See also Chapter XXIII.

The curves in Fig. 206 illustrate the importance of length. The diagram represents approximate curves of horse-power on base of speed for vessels of 9000 tons displacement of varying lengths, each having a coefficient of fineness of 50 per cent. We see from this the excessive horse-power required for the short ship as compared with the long ship of the same displacement when high speeds are reached. Indeed, a speed of 25 knots would be impossible for the 350 and 400 ft. ship, because of the excessive horse-power necessary.

The following table illustrates very forcibly the value of length when speeds are being reached which are high for the ship:—

Speed in knots . . .	10	12	14	16	18	20	22
Diadem—							
Ratio $\frac{V}{\sqrt{L}}$	0·48	0·58	0·67	0·77	0·86	0·96	1·06
I.H.P.	1,500	2,500	4,000	6,000	9,000	14,000	23,000
Powerful—							
Ratio $\frac{V}{\sqrt{L}}$	0·45	0·54	0·62	0·71	0·80	0·89	0·98
I.H.P.	1,800	3,100	5,000	7,500	11,000	15,500	23,000

The dimensions of these ships are as follows:—

Diadem, 435 ft. × 69 ft. × $24\frac{1}{2}$ ft. × 11,000 tons.

Powerful, 500 ft. × 71 ft. × $26\frac{1}{4}$ ft. × 14,200 tons.

The striking result is noticed that at 22 knots the I.H.P. required for the two ships of considerably different displacements would be the same, viz. 23,000 (the *Diadem's* engines are designed to indicate only 16,500 I.H.P), and if the curves of power on base of speed were drawn for the two ships, it would be noticed that for 23 knots the smaller ship would require more I.H.P.

When a model of a ship is run at the experimental tank to determine the resistance, it must be run at speeds *corresponding* to those of the ship. Such *corresponding speeds* are proportional to the *square root of the length*. Thus, if a ship is 500 ft. long and 23 knots, the speed at which a model 14 ft. long is run to determine the resistance must be $23\sqrt{\frac{14}{500}} = 3·85$ knots. Again, suppose we take as our model a ship 400 ft. long and 20 knots.

If we are designing a ship 500 ft. long of similar form, the speed corresponding to the 20 knots of the 400-ft. ship is $20\sqrt{\frac{500}{400}} = 22{\cdot}36$ knots.

Froude's Law of Comparison.—The resistances other than frictional of similar ships, or of a ship and her model at corresponding speeds, are connected by a most important law, called the *law of comparison*, viz.—

If the linear dimensions of a vessel be l times those of the model (or model ship), and the resistances of the latter at speeds V_1, V_2, V_3, etc., are R_1, R_2, R_3, etc., then at the corresponding speeds of the ship, $V_1\sqrt{l}$, $V_2\sqrt{l}$, $V_3\sqrt{l}$, etc., the resistances of the ship will be R_1l^3, R_2l^3, R_3l^3, etc.

The law of comparison would apply to the frictional resistance if this varied as the square of the speed, and if the coefficient were the same for long and short surfaces. Neither of these conditions are actually fulfilled, as we have seen, so the law cannot strictly be applied to frictional resistance.

In Mr. Froude's experiments on the *Greyhound*, mentioned above, experiments were also made with a model of the ship at the experimental tank to determine the resistance. It was found that, deducting the frictional resistance of the ship and the model from the total in either case, the remaining resistance of the ship compared with that of the model in accordance with the law of comparison as above stated. That is, the length of the model being one-sixteenth that of the ship, at speeds of the ship $\sqrt{16}$, or four times that of the model, the resistances of the ship other than frictional were practically 16^3, or 4096 times that of the model.

It is the established practice of the British Admiralty to have models made and run in the experimental tank in order to determine the resistance of the ships of the Royal Navy, and most valuable data is obtained in this way to determine the power necessary when considering new designs. Alternative forms can also be readily tried in order to determine the best possible form of underwater body for the desired speed. It may be mentioned that this best form cannot always be adopted, because of the conditions of stability which must necessarily be satisfied.

We have seen that E.H.P. $= \frac{1}{326}(R \times V)$, so that we can use E.H.P. in the law of comparison instead of resistance as follows :—

If E_1, E_2, E_3, etc., are the effective horse-powers (other than

frictional) at speeds V_1, V_2, V_3, etc., of the model (or model ship), then the E.H.P. of the ship whose linear dimensions are l times those of the model, at the corresponding speeds $V_1\sqrt{l}$, $V_2\sqrt{l}$, $V_3\sqrt{l}$, etc., are $(E_1 \times l^3 \times \sqrt{l})$, $(E_2 \times l^3 \times \sqrt{l})$, $(E_3 \times l^3 \times \sqrt{l})$, etc.

Methods of Estimating I.H.P.—There are a number of methods in vogue for estimating I.H.P.; the following are in use when an experimental tank is available:—

A model in paraffin wax is made at the tank and is towed at the series of speeds *corresponding* to those of the ship. From the resistances thus obtained a curve of E.H.P. on base of speeds can be drawn for the ship by using the law of comparison, after making the proper corrections for frictional resistance.

To convert the E.H.P. thus obtained into I.H.P., data obtained as to the value of the ratio $\frac{\text{E.H.P.}}{\text{I.H.P.}}$, or propulsive coefficient, in previous ships has to be used. As already stated, the usual value of this in ships of the Royal Navy is from 45 to 50 per cent. The value of the propulsive coefficient will not be constant for the same ship for all speeds, because the propellers are designed to be most efficient at the top speed, and the friction of the engines is relatively less when working at the highest revolutions.

In the early stages of the working out of a design, when it is not in a sufficiently settled state to have a model run, preliminary estimates may be made of power by using the information obtained from previous models or from the trial data of actual ships. The latter would have to be adopted if an experimental tank was not available.

The following example will illustrate how one would utilize the data already obtained from model experiments to make a preliminary estimate of power:—

Suppose it is desired to design a battle-ship of 12,000 tons of 20 knots speed. To get this speed one would need a fine form; a good breadth is necessary for stability purposes, and a moderate length is desirable for handiness. We have data concerning the E.H.P. of a vessel 320 ft. × 57 ft. × 19·5 ft. × 5150 tons, and this appears to have a form that is desirable. We therefore have the following calculations:—

$$\text{Ratio of displacement} = l^3 = \tfrac{12000}{5150} = 2{\cdot}33,$$
$$\text{so that } l = \sqrt[3]{2{\cdot}33} = 1{\cdot}325, \text{ and } \sqrt{l} = 1{\cdot}15.$$

The dimensions of the 12,000-ton ship would therefore be—

Length (320 × 1·325), breadth (57 × 1·325), draught (19·5 × 1·325), or, 425 ft. × 75½ ft. × 26 ft. × 12,000 tons, say.

The speed of the smaller ship *corresponding* to 20 knots of the 12,000-ton ship is $\frac{(20)}{(1\cdot15)} = 17\cdot4$ knots. At this speed the E.H.P. of the 5150-ton ship is found to be 2960, so that the E.H.P. of the larger ship at 20 knots is—

$$2960 \times 2\cdot33 \times 1\cdot15 = 7940.$$

If we assume a propulsive coefficient of $47\frac{1}{2}$ per cent., the I.H.P. would be $7940 \times \frac{100}{47\cdot5} = 16{,}750$ I.H.P.

We therefore have a ship—

425 ft. × $75\frac{1}{2}$ ft. × 26 ft. × 12,000 tons; 16,750 I.H.P., 20 knots.

(In the above calculation we have made no correction for frictional resistance, assuming it to vary according to the law of comparison. This gives the result for the larger ship a little in excess, so that we are somewhat on the safe side.)

The following example will illustrate how we have to proceed when using the trials of a previous ship as data :—

H.M.S. "Edgar" is 360 *ft.* × 60 *ft.* × $23\frac{3}{4}$ *ft.* × 7390 *tons, and when tried obtained* 10, 14, 18, 20 *knots with* 1000, 3000, 7500, 11,000 I.H.P. *A ship of similar form,* 11,000 *tons, is being designed, and it is desired to have a speed of* 21 *knots. Make an estimate of the power required.*

The ratio of displacement is $l^3 = \frac{(11{,}000)}{(7390)} = 1\cdot49$, so that ratio of linear dimensions is $\sqrt[3]{(1\cdot49)} = 1\cdot14$.

The new ship similar to *Edgar* is accordingly—

410 ft. × $68\frac{1}{2}$ ft. × 27 ft. × 11,000 tons.

The ratio of corresponding speeds is $\sqrt{1\cdot14} = 1\cdot07$, so that to 21 knots of new ship *corresponds* $\frac{(21)}{(1\cdot07)} = 19\cdot65$ knots of *Edgar*. By putting curve in to scale we find that 10,400 I.H.P. is required for *Edgar's* 19·65 knots.

For the new ship's 21 knots we therefore require $10{,}400 \times 1\cdot49 \times 1\cdot07 = 16{,}600$ I.H.P.

We tacitly assume that we shall obtain the same efficiency of propulsion as was obtained in the model ship.

Components of I.H.P.—We have seen above that of the power actually exerted at the engines of a vessel, only about one-half is usefully employed in overcoming the resistance experienced by the ship passing through the water. It will be of interest to note briefly the reasons for this difference.

The thrust necessary to overcome the resistance of a ship is developed by the projection sternward by the propeller of a column of water, and the reaction constitutes the forward thrust which is transmitted to the ship at the thrust block. A vessel moving at speed V, say, is accompanied by a belt of water drawn along by the friction of the ship's surface. This forward moving water is called the *frictional wake*. The motion of this wake at the stern is complex and variable, and gradually disappears as we go away from

the ship's surface, but we may regard it as equivalent to a uniform current of water having a forward velocity nV, say. The propeller, therefore, is not working in still water and passing through the water with speed V, but is working in water having a speed relative to the ship of $V - nV$. The propeller derives increased thrust from this cause, and the gain will be the greater for a single screw ship than for a twin screw ship, because the frictional wake is of greatest influence near the middle line, where the single screw is placed.

The resistance caused by the deflection of water at the forward part of a ship is nearly all returned to the ship in the forward thrust obtained by the closing in of the water at the stern. (In the ideal case of a body moving wholly immersed in a frictionless fluid, the net resistance in the direction of motion is zero.) Anything, therefore, which interferes with the natural closing in of the stream lines at the stern will be a cause of resistance. The presence of the propeller at the stern of a ship interferes in this way, and is a distinct cause of resistance. This *augment of resistance,* as it is termed, will be greater for a single screw ship than for a twin screw ship, because in the latter case the screws are further away from the hull.

We notice that in a single screw ship, although the gain due to the frictional wake is greater than in a twin screw ship, yet the loss due to augment of resistance is also greater, and it is found that under ordinary circumstances the gain due to wake is practically equal to the loss due to the augment of resistance in both single and twin screw ships.

The various items which make up the I.H.P. may be stated approximately as follows, in the case of a twin screw vessel with fast-running engines of high pressure:—

	Per cent.
Dead load friction, due to dead weight of working parts, friction of packings, bearings, etc.; approximately constant at all speeds	7
Working load friction, varying with thrust of propeller and speed of engines	7
Working of air pump off the main engines . . .	1
Loss at propeller, due to slip, friction of blades and augmentation of resistance, allowing for the gain due to the frictional wake	33
Effective horse-power expended in overcoming the net resistance of the ship to the onward motion . . .	52
	100

In the above we see that 15 per cent. of the power is expended in the machinery before it reaches the propeller. This will be exceeded at low speeds, because then the dead load friction is relatively of greater importance. Also, seeing that the propeller is designed to be most efficient at the highest speed, it is evident from these two causes that the propulsive coefficient will be less at the lower speeds than at the higher speeds.

Progressive Speed Trials.—The trials required to satisfy contract conditions of ships of the Royal Navy are (except for destroyers, etc.) to determine *power* and not *speed*. In the design of a ship a certain power is specified, and if this power is obtained for the given length of time and under the specified conditions, the failure of a ship to obtain the speed would be the responsibility of the designer and not of the contractor.

In order to determine the real speed and to obtain data for future use, it is necessary to run trials on the measured mile or over a course of known length. This is not always done with every ship, but usually with at least one ship of each class, and for this ship a series of progressive trials is run at varying speeds in order to obtain a complete curve of power on base of speed, as *e.g.* H.M.S. *Drake* (Fig. 205). For these speed trials there can obviously be only one standard, viz. with the *highest possible efficiency*. The bottom should be clean, the weather favourable, the engines and boilers in perfect order, the best coal and most skilful supervision and stoking. If any of these conditions are not satisfied, then we can have no scientific knowledge of the performance, as the influence of the various factors affecting the ship is unknown. It is, however, not unusual to find in the history of a ship, subsequent to her trials, that, when the engines have got into perfect working order, the ship has done better than even when tried under the conditions above mentioned.

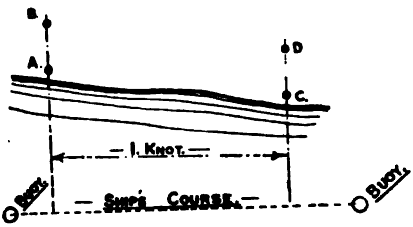


Fig. 207.

In running speed trials a measured mile course is usually used, on which, after careful survey, two pairs of posts, AB and CD, have been placed, as Fig. 207, exactly a nautical mile (6080 ft.) apart. Two buoys mark a course at right angles to the posts, as shown (or the course is on a given

bearing, as "due north" in one case). The ship is brought on the course, and the time of passing each pair of posts is taken by a chronometer stop watch. There are a number of points to be observed in connection with these measured-mile runs, viz.—

1. *Measurement of power.*—This is done by the engine-room staff. Indicator cards are taken from each piston showing how the pressure of the steam varies at each point of a whole revolution. A calculation from these cards enables the I.H.P. to be determined for each run. In the case of very fast-running engines, however, the time in which an indicator card is taken is so small that a small error in the tracing of the steam line will mean a large error when converted into horse-power.

2. *Noting the time of traversing the mile.*—Here there may be errors due to—

(*a*) Indefiniteness of the mile posts; and
(*b*) Taking the time.

In the former the "personal equation" comes in, as in astronomical observations. One observer will have a tendency to anticipate, while another will delay starting or stopping the stop watch until he is perfectly sure that the posts are coincident. Several observers usually take observations, so that the mean may be taken if any doubt occurs.

In the latter the error involved should be small. If, however, the vessel is of high speed, a small error in timing means a considerable error in the speed recorded. Thus, half a second means $\frac{1}{8}$ knot at 30 knots.

3. *Growth of tide.*—If the tide were uniform in speed and direction the mean of any two or any even number of runs, half in one and half in the other direction, would give the true speed of the ship. As a matter of fact, however, the tide varies in speed. At Stokes Bay, for instance, the following are the speeds of tide at quarter-hour intervals as found by Mr. Froude, viz. 0·13, 0·39, 0·66, 0·89, 1·11, 1·33 knots, etc. Thus a vessel whose true mean speed was 15 knots, if run at quarter-hour intervals, would travel 15·13, 14·61, 15·66, 14·11 knots, and the ordinary mean of these speeds would show only 14·88 knots. In practice the error due to this cause is minimized by taking the mean of means, thus :—

Speeds.	1st Means.	2nd Means.	Mean of Means.
15·13			
	14·87		
14·61		15·002	
	15·135		15·006 knots.
15·66		15·01	
	14·885		
14·11			

4. The ship on entering the measured-mile course must have attained a uniform speed; if this is not so the ship is being accelerated, and this acceleration is a cause of resistance which will have its effect on the speed obtained.

5. *Running in shallow water.*—Anything which interferes with the natural paths of the streams of water round a ship is a cause of resistance. In shallow water the distance between the ship and the bottom of the water is not sufficient to allow the natural stream lines to be formed, and they become restricted and cause an increase of resistance. A conspicuous instance of this was noticed in the trials of H.M.S. *Edgar*. When tried at Stokes Bay, with a depth of water of 12 fathoms, 13,260 I.H.P was required for 20½ knots. When tried on the deep-sea course between Plymouth and Falmouth, 21 knots was obtained with 12,550 I.H.P., or about ¾ knot difference for the same power.[1]

In consequence of this the trials at the higher speeds of large ships of the Royal Navy cannot be run at the measured mile at Stokes Bay, and it has been usual to run them on the deep-sea course between Rame Head and Dodman Point (29 fathoms), or at Chesil Beach (17 fathoms). The finest course for running these trials is at Skelmorlie, near the Clyde, where the depth of water is over 40 fathoms, and the course is sheltered. This course is used whenever possible. It has frequently been observed, however, that vessels of abnormally high speed do better in very shallow water than in deep water. (For a discussion of this interesting phenomenon reference must be made to more advanced treatises on the subject.)

[1] "My attention was very forcibly drawn to the importance of this matter in 1886, when a cruiser of about 19 knot speed which we had built obtained a greater speed after she had left our hands than had been obtained on the measured mile. . . . In looking for the reason it occurred to me that it might be due to the shallowness of the water over the course, and in measured-mile trials afterwards made this was shown to be the case."—Mr. Philip Watts, F.R.S. (I.N.A., 1892).

In running progressive trials it is necessary to have the ship as near to the normal load draught as possible. When a ship is deep or light the horse-power required for a given speed will vary approximately as the displacement.

From the information obtained on such a series of trials a curve of I.H.P. on base of speed can be constructed, as Fig 205, and then it can be at once determined what I.H.P. is required for a given speed, or what speed can be got for a given I.H.P under trial conditions. The curve of revolutions on base of speed can also be drawn as shown, and such a curve, showing revolutions necessary for any desired speed, is found very useful in the subsequent history of the ship. Records of all trials are kept in the ship's book for the information of the officers commanding.

Power Trials.—The following series of trials are now carried out in ships of the Royal Navy.

1. *Battle-ships, first class cruisers, second class cruisers.*
    (*a*) Preliminary trial at sea.
    (*b*) 30-hour trial at about one-fifth full power.
    (*c*) 30-hour trial at about 70 to 75 per cent. full power.
    (*d*) 8-hour trial at full power.
    (*e*) Trial after opening up (24 hours).
2. *Third class cruisers.*
    (*a*) Preliminary trial at sea.
    (*b*) 30-hour trial at half forced draught power.
    (*c*) 8-hour trial at authorized natural draught power.
    (*d*) 4-hour trial at full forced draught power.
    (*e*) Trial after opening up (24 hours).

It is important to note the conditions under which the machinery and boilers are designed to exert the maximum power. It is for a certain limited time only in each case. The *Drake*, for instance, is not a $23\frac{1}{4}$-knot ship in the same sense as the Atlantic liner *Kaiser Wilhelm der Grosse* is a $22\frac{3}{4}$-knot ship. The I.H.P. in each case is about the same, viz. 30,000, but in the latter case the machinery is designed to maintain the speed across the Atlantic, and the weight involved is nearly double that for the *Drake* of the same power. In fact, such fast liners are practically only able to carry themselves and the necessary coals across, the amount of deadweight cargo they can carry being very small. The *Drake*, on the other hand, carries a large weight of armour and armament, but the machinery is only intended to attain 30,000 I.H.P. for a period not exceeding 8 hours. In a vessel like *Pioneer*, for instance

the full power is 7000, but this is obtained by forcing the boilers, and cannot be maintained for a longer period than 4 hours; the authorized natural draught power is 5000, and this is intended for a period not exceeding 8 hours.

In dealing with ships of the Royal Navy the following terms are in use, viz.—

(*a*) The authorized natural draught power is taken as the unit.

(*b*) "With all despatch," four-fifths the unit, for 30 hours (this is when great urgency is necessary).

(*c*) "With despatch," three-fifths the unit; this should not be materially exceeded when the period of steaming exceeds 30 hours.

(*d*) "With moderate despatch," two-fifths the unit.

(*e*) Ordinary speed, one-fifth the unit.

The maximum speed at which the *Drake*, for instance, could proceed so long as her coal lasted would be about 21 knots, corresponding to 18,000 I.H.P. The speed at starting, with bunkers full, would be something less than this, but it would get greater as the ship lightened.

CHAPTER XXIII.

THE DESIGN OF WAR-SHIPS.

THE service for which a ship is intended to be employed has manifestly the predominating influence on her design. The duties which ships of a navy like the British Navy have to perform are so varied, that no single ship could possibly combine all the qualities needed in war-ships. Thus very high speed, heavy armour protection and powerful armament cannot all be embodied in one design. A compromise is necessarily effected, and if we sacrifice some protection and guns to obtain high speed and large coal capacity we have a *cruiser*; if we have less speed and pay most attention to protection and armament we have a *battle-ship*. If we want a ship that shall be able to keep the sea for long periods without docking, we must have a vessel *sheathed* with wood and copper, and in doing so we have to accept some increase of cost and decrease in measured-mile speed, as compared with a vessel with an ordinary steel skin. Again, a vessel intended for coast defence would need only a moderate coal capacity and a small draught of water. There are navies in which such a type of ship would be valuable; the construction, however, of battle-ships of small size has been discontinued in the British Navy for some years.

The design of a war-ship would be an almost impossible task apart from experience and data obtained from previous ships. When the main features which it is desired to embody in a new design are given by the authorities, it is the function of the naval architect to work out such a design as shall satisfactorily embody those features. Experience in the specialities of war-ship design is a necessary qualification, as the conditions to be satisfied are altogether different to those in the case of merchant steamers.

There are many qualities which to a greater or less extent must be found in any war-ship design. Some of these are—

1. *Strength,* both structural and local. We have already discussed this at some length.

2. *Stability.*—This is a vital quality. A war-ship must have sufficient stability left after sustaining a reasonable amount of damage. It is on this account that the metacentric heights given to war-ships are greater than obtain in merchant steamers. The stability at large angles also requires careful consideration, because of the high position of the C.G. of ship. The question of the most economical propulsion frequently has to go into the second place in order to obtain a proper amount of stability.

3. *Speed.*—This depends on the intended service of the ship. Speeds have considerably increased during recent years, this having been rendered possible by the use of watertube boilers, with high steam pressures and high revolutions.

4. *Handiness.*—The influence of the shape of the stern and the rudder on turning have already been discussed. A short ship also is handier than a long ship, other things being the same.

5. *Habitability.*—This is important because of the necessity of keeping the crew in a good state of health. A high freeboard ship has a great advantage over a low freeboard ship in this respect, the living spaces being much more airy and light.

6. *Convenient transport of coal and ammunition.*

7. *Economy of first cost and maintenance.*—These two things are sometimes opposed. Thus a steel ship will be cheaper than a sheathed ship because of the cost of the sheathing and the metal stem, etc., necessary. The cost of maintenance of the sheathed ship, however, will be considerably less than the steel ship, because it will not foul so quickly or require such frequent docking.

8. *Length of vitality.*—The amount of coal, ammunition, etc., carried by a ship will determine how long she can remain efficient as a fighting machine. The coal will determine the *radius of action.*

9. *Slowness of destruction.*—This includes protection by armour and decks, and the provision of minute subdivision.

10. *Armament.*—Being the available provision for attack—guns, torpedo equipment, ram.

Of the first stages of a design, Sir William White says ("Manual of Naval Architecture")—

"In the preliminary stages the processes are necessarily tentative and subject to correction. The various features of the design are, to a large extent, interdependent. At the outset the dimensions, form, and displacement are undetermined. Yet upon them depend the power which the engines must develop to give the desired speed, the weight of the hull, and the weight of certain parts of the equipment. In the finished ship the sum of the weights of the hull structure, propelling apparatus, equipment, coals, and load must equal the displacement to

the specified load-line. Apart from experience, a problem involving so many unknown yet related quantities could scarcely be solved. On the basis of experience, recorded data, and model experiments it is dealt with readily. Approximate dimensions and forms are first assumed. The weight of hull is then approximated to for the system of construction adopted and the type of ship. An estimate of the probable engine power is made, either on data obtained from the steam trials of previous ships or from model experiments. The weight of the engines and boilers is then ascertained for the horse-power, and the rate of coal consumption per hour calculated on the same basis, while the total weight of coal for the intended steaming distance at the desired speed is readily deduced. Adding together these first approximations to the weights of hull, equipment, machinery, and coal, and to the total adding the load stipulated to be carried, a grand total is reached which should equal the displacement provisionally assumed. If the sum total is in excess or defect of the provisional displacement, corrections must be made on the dimensions originally assumed, with a view of obtaining a balance. For these corrected dimensions a fresh series of approximations is made to the weights of hull, equipment, machinery, and coal. A balance between the grand total of weights, and the displacement corresponding to the form and dimensions, is ultimately obtained. When no large departure from previous experience or precedent is made, this preliminary work is rapidly performed. Under other circumstances, the selection of the most suitable dimensions and form may involve the consideration of many alternatives."

The total displacement of a completed design is made up of the following items, viz.—

1. General equipment.
2. Armament.
3. Machinery.
4. Engineer's stores.
5. Coal.
6. Armour and protection to hull.
 " " armament.
7. Hull, including structure and fittings.
8. Board margin.

1. *General equipment.*—This includes fresh water; provisions (including bread and spirits); officers' stores (including ward-room and gun-room stores and paymaster's slops); officers, men, and effects; anchors; cables; masts, rigging, etc.; boats; warrant-officer's stores. These weights depend largely on the type of ship and on the complement. The intended service of the ship has an influence, as the weight of stores allowed would be greater or less according as the vessel is intended for distant, isolated service, or with a fleet not far from a dockyard.

2. *Armament.*—The weight of this can be very closely estimated

when the detail of the armament is settled. An important point in connection with this is the number of rounds taken per gun.

3. *Machinery.*—When the I.H.P. is provisionally settled,[1] an estimate is prepared by the Engineers of the necessary weight. This will depend on several things, *e.g.* the type or types of boiler to be used; the revolutions and stroke of the engines and the speed of the pistons; the degree to which the boilers are to be forced. For the largest set of engines fitted up to date (15,000 I.H.P.), 120 revolutions is the maximum; as engines of lower power are reached higher revolutions are possible; thus the engine for 6250 I.H.P. has 180 revolutions, and for 4900 I.H.P. has 250 revolutions. Destroyer engines are faster still. The adoption of watertube boilers has had a great influence on design in recent years, enabling a larger power to be developed on a given weight than formerly. The employment of turbine machinery will doubtless greatly influence the conditions of design in the near future.

4. *Engineer's stores.*—The allowance of these stores depends on the intended service of the ship as well as on the power.

5. *Coals.*—It is the practice in the designed displacement of H.M. ships to include a certain weight of coal. This is called the legend weight. Thus the *Royal Sovereign* has 900 tons, as also *Majestic*, *Formidable*, and *Duncan* classes. The total capacity available for coal is considerably more than this, being over 2000 tons in the latest battle-ships. All the official steam trials to test the speed are carried out at the draught corresponding to the *legend* condition. Sometimes, however, trials are carried out before the completion of the ship to determine the acceptance of machinery from the contractors. These trials are simply for I.H.P., and not for speed, and then the ship is not necessarily ballasted to her normal load line, so long as proper immersion is given to the propellers.

6. *Armour and deck protection.*—The weight devoted to protection may be divided into—

(*a*) Vertical armour for the protection of the buoyancy and stability;

(*b*) Vertical armour for the protection of the armament; and

(*c*) Deck protection.

The percentage of each of these of the total weight of protection in a recent battle-ship was as follows: (*a*) 38 per cent.; (*b*) 34 per

[1] See Chapter XXII.

cent.; (*c*) 28 per cent. This shows clearly how large a proportion is devoted to the effective protection of the armament. This includes barbettes and casemates, but not gun shields, which are taken in the armament. The barbettes especially are well protected. Thus in the *Duncan* the side armour is 7 in., but the barbette armour is 11 in. The reason is that each barbette represents such a large proportion of the fighting power of the ship. A single shot piercing the belt might not be a serious matter, but one shot through the barbette armour would probably cripple nearly one-half of the ship's fighting capacity.

A feature of modern battle-ship designs has been the larger area covered with armour than formerly, with a corresponding reduction of thickness. This has been fully dealt with in Chapter XIII. The improvements made in the quality of armour has also had a great influence on cruiser designs. Up to the *Diadem* the protection was considered to be best obtained by a thick protective deck at the waterline, as Figs. 21 and 22. Owing, however, to the introduction of Krupp armour, this system of protection was modified by the adoption of a broad patch of 6-in. armour in the *Cressy*, over about half the length, in association with thick decks and bow protection (see Figs. 136 and 137). Armour protection has been adopted up to the present time for first class cruisers.

7. *Hull.*—The weight devoted to the hull comprises—

(*a*) Weight of the structure; and

(*b*) Weight of fittings, etc., not contributing to the structural strength.

The total weight of hull in large ships varies from 35 to 40 per cent. of the total displacement, and it is only by a most careful arrangement of the material, combined with high-class workmanship, that the weight can be brought as low as this. The corresponding weight in large merchant steamers is considerably in excess of the above figure; this is very notable when we consider that about one-half of the weight of hull in a war-ship is concerned with fittings, etc., which do not contribute primarily to the structural strength.

Economies of weight during building are important to keep weights down, and a great deal can be done in this direction with no loss of strength. Lightening holes are largely employed. Thus in Fig. 16, showing a longitudinal, we notice that the plate must be weakened by holes connecting it to the transverse frame, so that we can well afford to cut away the plate by a hole betwee-

the frames without causing any reduction in strength. The lightened liner, shown in Fig. 47, is also an instance of reduction of weight with no reduction of strength. Such small savings of weight do not appear to be in themselves of much value, but in the aggregate they amount to a considerable saving of weight.

Considerable reductions of weight have been effected in recent ships by lessening the duplication, etc., formerly fitted. One instance of this is seen in the abolition of relieving tackles to the steering gear. The stockless anchors now adopted have enabled the large weight formerly devoted to catheads, bill-boards, etc., to be saved. A large amount of weight has been saved in recent ships by the omission of teak linings and light boxes to magazines, omission of wood decks, simplification of pumping arrangements, etc.

8. *Board margin.*—This is a weight provided for at the time of the design to cover alterations or additions made during the progress of the building of the ship. Any weight thus required, not provided for in the original legend of weights, has to be taken out of the Board margin, and specially submitted for approval to the Board of Admiralty.

Influence of Weight saved or added on a Design.[1]—It is worth noting that a weight saved in any way has an influence on a design far greater than is given by the number of tons thus saved. Thus, suppose in any part of a design, say the equipment, 50 tons can be saved. The influence of this 50 tons less is felt in all parts of the design. The ship thus lightened requires less I.H.P. for the same speed; the engines, etc., thus weigh less and require a smaller complement. A smaller-sized ship will then be sufficient, which will weigh less than before and require less I.H.P. These things thus act and react upon one another, and, as a final result, we should find that by saving 50 tons on the equipment a saving on the whole design, amounting to 100 tons or more, would be possible.

As an extreme case, it is calculated that the adoption of two 56-ft. steam pinnaces (each 18 tons), now used instead of the two 37-ft. steam pinnaces (each 9 tons) formerly supplied, has caused a weight of about 150 tons to be added to the total weight of the design of large ships. The actual additional weight of the boats carried is only 18 tons. But the heavier boats require strong

[1] See a paper on "The effect of modern accessories on the size and cost of warships," by Mr. Whiting, Assistant-Director of Naval Construction (I.N.A., 1903).

masts, derricks, and steam or hydraulic hoists for lifting, and special stowage; so that the result is an additional weight of 70 tons to be carried about. The influence of this is to necessitate a design to be about 150 tons heavier than would have been necessary had the lighter boats been carried.

Stability of a Design.—Besides fixing on the legend of weights, we need concurrently to make a calculation for the C.G., both in a vertical and horizonal direction. The first is required to obtain sufficient stability, and the second to see that the vessel when complete shall float at the desired trim.

The type of calculation, as prepared for a small cruiser, is shown in the following table:—

Items.	Tons.	From L.W.L.				From mid-length.			
		Below.		Above.		Before.		Abaft.	
		Lever.	Moment.	Lever.	Moment.	Lever.	Moment.	Lever.	Moment.
General equipment—									
Water	25	4·0	100	—	—	—	—	12·0	300
Provisions	30	4·5	135	—	—	25·0	750	—	—
Officers' stores	15	2·0	30	—	—	—	—	125·0	1,875
Officers, men, and effects	30	—	—	6·0	180	55·0	1,650	—	—
Cables	30	4·0	120	—	—	85·0	2,550	—	—
Anchors	10	—	—	15·0	150	90·0	900	—	—
Masts, etc.	25	—	—	45·0	1,125	—	—	7·0	175
Boats	10	—	—	21·0	210	—	—	20·0	200
Warrant-officer's stores	20	1·5	30	—	—	65·0	1,300	—	—
Armament	175	—	—	4·0	700	—	—	5·0	875
Machinery	450	4·0	1,800	—	—	—	—	33·0	14,850
Engineer's stores	50	0·5	25	—	—	—	—	70·0	3,500
Coals	300	0·2	60	—	—	3·0	900	—	—
Protective deck	210	—	—	1·5	315	—	—	15·0	3,150
Hull	1250	—	—	1·5	1,875	—	—	11·5	14,375
Margin	20	—	—	5·0	100	—	—	—	—
	2,650		2,300		4,655		8,050		39,300
					2,300				8,050
					2650)2,355				2650)31,250
					0·89 ft. above L.W.L.				11·8 ft. abaft mid-length.

Total weight, 2650 tons.
C.G. above L.W.L., 0·89 ft.
Transverse metacentre above L.W.L., 2·9 ft.
Metacentric height = 2·9 − 0·89 = 2 ft.

Thus the estimated metacentric height in the legend condition is 2 ft., and the form of the ship must be made to displace 2650 tons, with the centre of buoyancy 11·8 ft. abaft mid-length, in order that the vessel may float at the required draught and trim. Also, the transverse metacentre must be 2·9 ft. above L.W.L. to get a metacentric height of 2 ft.

This determines the initial stability. Calculations must, however, be made to determine the "curve of stability," to see that the ship has sufficient stability at large angles of inclination.

The detailed calculation of the weight and position of the C.G. of hull is a long and complicated operation, and although this is usually done as the design proceeds, yet in the early stages it is necessary to make an approximation. For this the information obtainable from the inclining experiments of previous ships is invaluable. It is the practice now to incline the ships of the Navy, not only to ascertain the stability of the ships themselves, but also to afford data for future designs. The D. 284 form filled up by the dockyard, and the D. 211 form filled up by the ship's officers, are also of extreme value in affording information as to weights to the designing staff at the Admiralty.

The tendency of modern designs is to lead to increased weight above, by the adoption of an armoured battery and heavier guns on the upper deck, protected by heavy shields. This tendency necessitates the stability being carefully considered, because of the higher position of the C.G. thus caused. The influence of this is seen, for example, in the breadth given to the *Duke of Edinburgh* as compared with the *Drake* of greater displacement.

Drake, 500 ft. × 71 ft. × 26 ft. × 14,100 tons; 2 9·2-in. guns with shields; 16 6-in. guns in casemates.

Duke of Edinburgh, 480 ft. × 73½ ft. × 27 ft. × 13,550 tons; 6 9·2-in. guns with shields; 10 6-in. guns in battery.

It will be remembered that in Chapter XVII. we saw that the breadth of ship has a great influence on the position of the transverse metacentre. In a ship where the conditions of the design lead to a high C.G., the transverse metacentre must also be high to get sufficient metacentric height.

Horse-power and Speed.—The methods adopted to obtain an estimate of the horse-power necessary for any desired speed have been dealt with in Chapter XXII. The length of a ship in relation to the speed to be attained has a most important influence on economical propulsion. To say that a ship has a speed of

15 knots, say, does not convey any correct idea as to whether the speed is high or not for the ship, unless it is coupled with the size or length of the ship. Fifteen knots would be a high speed for a vessel 150 ft. long, but quite a moderate speed for a vessel 500 ft. long. A measure of speed is obtained by comparing it with the *square root of the length.* When this ratio $\frac{V}{\sqrt{L}}$ is *above unity* we have a speed which is high for the ship, and which requires a very large expenditure of horse-power to obtain. It is interesting to note that, in the Atlantic liners, as speeds have gone up, so lengths have increased, keeping the ratio $\frac{V}{\sqrt{L}}$ nearly constant. Thus we have—

Ship.	Speed in knots.	Length in feet.	Ratio $\frac{V}{\sqrt{L}}$
Etruria	19·5	500	0·87
Teutonic	20	566	0·84
Campania	22	600	0·9
Deutschland	23½	666	0·9
New Cunarders . .	25	760	0·91

When, however, we come to cruisers, we find that we have speeds which are much higher, relative to the size of ships. Thus we have—

Ship.	Speed in knots designed.	Length in feet.	Ratio $\frac{V}{\sqrt{L}}$
Edgar	20·5	360	1·08
Drake	23	500	1·03
Monmouth	23	440	1·1
Pelorus	20	300	1·15

As regards economy of propulsion alone it would doubtless be better to increase the length of cruisers, but it is not desirable to make them longer than absolutely necessary, as the longer ship requires heavier scantlings and more protection, and affords a larger target than the shorter ship, besides being less handy in turning. The gain in propulsion obtained by the longer ship would be more than lost in other directions.

"In some quarters, whatever length is decided upon, it is pronounced to be 50 ft. too short, as a rule, without any investigation of what such an addition would involve. Where criticisms have been associated with alternative proposals —and such cases are few—it has been the writer's task to investigate them. *In no single case* so treated has it appeared that the proposals made would have given the gains in propulsion anticipated, in association with other supposed advantages. As a rule, the proposals made have been proved to be incompatible with a due provision of stability." [1]

Comparison between "Juno" and "Hyacinth."—A comparison between these two ships is interesting, because they have the same dimensions and displacement, but the weight is disposed differently, giving very different qualities.

	Juno.	*Hyacinth.*
Dimensions	350 ft. × 54 ft. × 20 ft. 6 in. × 5,600 tons	
I.H.P. (natural draught)	8,000 (8 hours)	10,000 (8 hours)
Corresponding speed	18 knots	20 knots
I.H.P. (forced draught)	9,600 (4 hours)	—
Corresponding speed	19½ knots	—
Armament	5 6-in. 6 4·7-in.	11 6-in.

Thus the *Hyacinth* got considerably more both in speed and armament than the *Juno* on the same displacement. The reason of the difference is found in the different types of machinery. The *Juno* had cylindrical boilers of 155-lb. pressure, with engines of 39-in. stroke and 140 revolutions. The *Hyacinth* had watertube boilers of 300-lb. pressure, with engines of 30-in. stroke and 180 revolutions.

The machinery of the latter ship, developing 25 per cent. more power at natural draught, was obtained on rather less weight than in the *Juno* on account of the watertube boilers, higher pressure and quicker running engines. The smaller height of the engines also made it possible to get them beneath the protective deck, and so the armour protection to the cylinders, necessary in the *Juno*, was saved in the *Hyacinth*. The weight thus made available was used to replace the 6 4·7-in. guns of the *Juno* by 6 6-in. guns in the *Hyacinth*, and thus a ship of much greater power as a fighting machine was obtained.

Comparison between "Topaze" and "Sentinel."—An example of two recent designs having nearly the same displacement,

[1] Sir William White, K.C.B., *Cassier's Magazine*, August, 1897.

but of different qualities, is seen in the third class cruiser *Topaze* and the scout *Sentinel*. The following table shows the main features of these vessels :—

	Topaze.	*Sentinel.*
Length	360 ft.	360 ft.
Breadth	40 ft.	40 ft.
Draught	14·5 ft.	14·25 ft.
Displacement	3,000 tons	2,900 tons
I.H.P.	9,800	17,000
Speed	21¾ knots	25 knots
Armament	12 4-in. 8 3-pounders 2 torpedo-tubes	10 12-pounders 8 1¼-pounders 2 torpedo-tubes
Protection	deck, 1-in. flat 2-in. slope	deck, ⅝-in. flat 1½-in. slope
Coals carried at above draught	300 tons	150 tons

In the latter case it is seen that the ship has only a slight armament of 12-pounder and smaller guns, with only 150 tons of coal at the designed draught. The machinery is 17,000 I.H.P. for the 25 knots desired. Thus the ship is only able to carry herself and her machinery with a comparatively small load, in order to reach the high speed of 25 knots. In the former case the ship is able to carry an armament of 4-in. guns, with a large amount of coal, but this increase of load carried can only be obtained by having the lower speed of 21¾ knots. It is seen that an increase of speed from 21¾ to 25 knots, or an increase of 15 per cent., means an increase of nearly 75 per cent. on the power.

CHAPTER XXIV.

NOTES[1] *ON THE LOSS OF H.M.S. "VICTORIA"* (June 22, 1893).

CIRCUMSTANCES leading up to the collision:—

The fleet was proceeding in two lines, the *Victoria* leading the starboard column, the *Camperdown* leading the port column. The ships were proceeding at a speed of 8·8 knots, the two lines being 1200 yards apart. At the time of the signal to turn, the helm of *Victoria* went "hard-a-starboard" 35° (corresponding to a tactical diameter of 600 yards). The helm of *Camperdown* was put at 28° (corresponding to a tactical diameter of 800 yards). When the ships had turned through eight points, it was recognized that collision was inevitable, and the port engines of *Victoria* and the starboard engines of *Camperdown* were reversed, but this had little effect, as the collision took place one minute later. The speed of the ships at the time of the collision was from 5 to 6 knots.

The *Camperdown* struck the *Victoria* nearly at right angles, about 65 ft. abaft the stem. The blow was just before important transverse bulkheads (Fig. 208). Both the ships turning rapidly at the time of the collision caused the sterns to swing together, and this considerably widened the breach in the side of the *Victoria*. This, together with the hole caused by the original blow, destroyed the connections of the bulkheads above-mentioned with the side of the ship. The value of these bulkheads was thus completely destroyed, and the compartments on either side were thrown open to the sea.

For the first minute after the collision the two vessels were locked, and during this time the *Victoria* heeled slightly to starboard and settled a little by the bow. After the *Camperdown* had cleared, the *Victoria* continued to settle by the bow and to increase her heel to starboard. These movements proceeded gradually for

[1] See also end of Chapter VI. regarding the question of watertight doors.

about ten minutes, when a sudden lurch to starboard took place. The ship turned bottom up and finally sank by the head.

It will be desirable to briefly describe the main features of the *Victoria* in so far as they bear on this question. The ship was built to carry two 111-ton guns in a turret forward, and

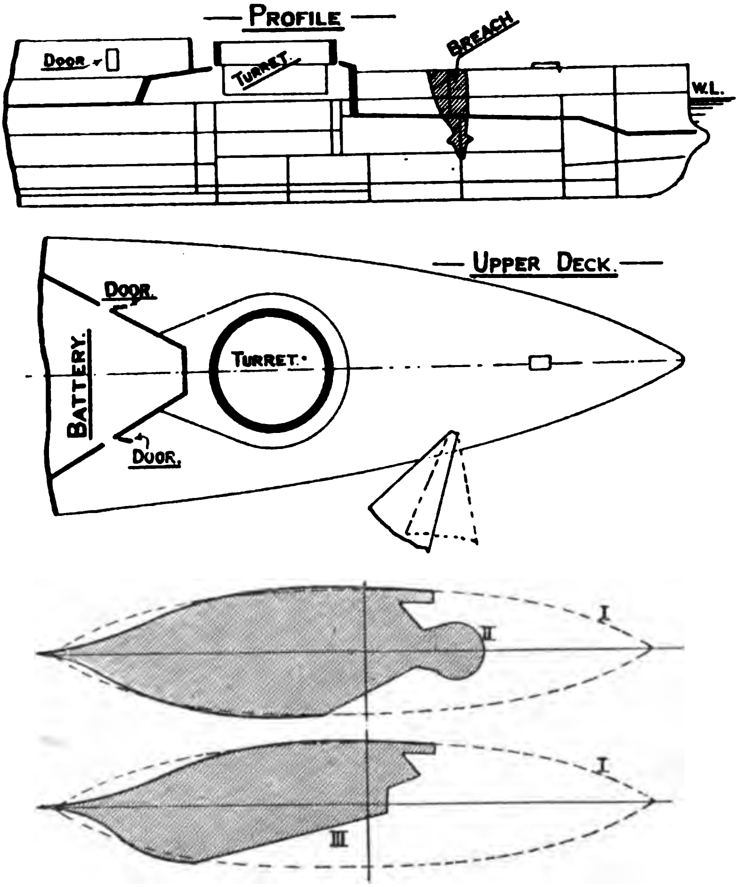


FIG. 208.

the great weight of this armament and the armour necessitated a somewhat low freeboard forward, viz. about 11 ft. Abaft the turret was a battery with fronts inclined abaft the beam, as shown, to give a large arc of training to the heavy guns. These fronts had armour doors in them, and the sides of the battery

had ports for the 6-in. guns. With these doors and ports closed, the freeboard near midships was about $18\frac{1}{2}$ ft., with them open the freeboard was about 12 ft.

At the time of the collision a number of watertight doors forward were open, the turret ports and scuttle on upper deck

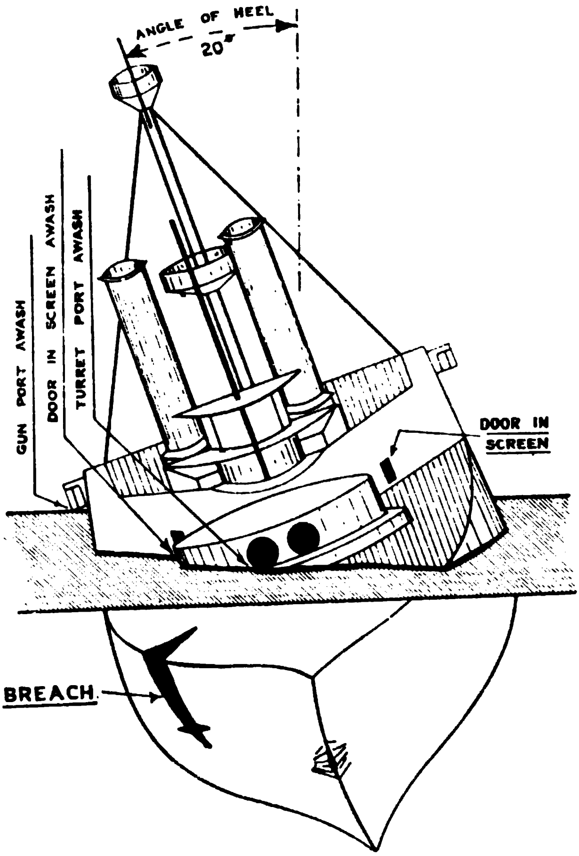


FIG. 209.

were open, and the doors in the front of the battery and the gun-ports in battery side were also open.

For convenience it will be desirable to consider separately the two movements which actually proceeded simultaneously (Fig. 209), viz.—

1. Depression of the bow.

2. Heel to the wounded (starboard) side.

1. *Depression of the bow.*—About four minutes after the collision the bow had sunk enough to bring the stem-head under water The forward part of the deck then became submerged, allowing water to pass down the scuttle on the upper deck and then down the turret ports, the water level reaching to the sill of the door in the battery front starboard, and the bottom of the foremost gun-port. At this moment the stem-head was depressed about 23 ft. from its normal position.

2. *Heel to starboard.*—Accompanying this depression by the bow, a gradual heel to starboard took place, until a heel of about 20° was reached. At this time the door in the battery front and the foremost 6-in. gun-port were just awash.

Both the above movements were accelerated by the motion ahead of the *Victoria*, the ship being made to steam slowly towards the land with the helm hard-a-starboard.

At this time, at the heel of 20°, a *sudden lurch to starboard* took place, and the vessel capsized and went down head first.

Calculations have been made, using for data the observed conditions, to investigate the cause of the sudden lurch that was observed; the suddenness of this lurch was without doubt the cause of the great loss of life.

The loss of buoyancy caused by the opening up of the compartments forward, which were inevitably flooded by the collision, and those flooded subsequently through open doors and hatchways, caused a change of trim of 29 ft., or 23 ft. depression forward and 6 ft. lift of the stern. Taking the ship in this condition, but supposing the turret ports, battery doors, and gun-ports *closed*, the vessel would have a positive metacentric height of $\frac{8}{10}$ ft. In Fig. 208 the shape of the intact waterplane is shown by I, and the shape of the waterplane area under the above condition is shown by II. In Chapter XVI. it is seen that the distance between the centre of buoyancy and the transverse metacentre is directly proportional to the transverse moment of inertia of the waterplane at which the ship is floating $\left(\mathrm{BM} = \frac{\mathrm{I}}{\mathrm{V}}\right)$. Under the conditions assumed above, the waterplane area would have a transverse moment of inertia of 3,888,000, giving such a position of the transverse metacentre that a metacentric height of $\frac{8}{10}$ ft. was retained. The ship under this assumed condition was in a condition of *stable equilibrium.*

Taking now the actual state of the ship with the turret ports, the battery doors, and the gun-ports *open*, the waterplane would be *suddenly* reduced to the shape shown by III in Fig. 208. This area only has a transverse moment of inertia of 2,783,000, lowering the transverse metacentre and bringing it below the C.G. This gave the ship a negative metacentric height of 1·8 ft., which rendered the ship *unstable*, and so she capsized.

Condition of ship.	Transverse moment of inertia of waterplane.	Metacentric height.
I. Intact and upright	6,023,000	5·05 ft.
II. At trim and heel observed just before lurch, with water excluded from turret and battery	3,888,000	0·8 ft.
III. At trim and heel observed just before lurch, but turret and battery flooded .	2,783,000	− 1·8 ft.

The fore-and-aft distribution of the waterplane area, shown in Fig. 208, made a great reduction in the fore-and-aft moment of inertia of the waterplane, and this considerably decreased the "moment to change trim 1 in.," rendering the ship less and less able to resist change of trim as the water gained access to the forward compartments.

The following conclusions were reached as the result of the inquiry into the circumstances of the *Victoria's* loss:—

1. So far as can be judged, had all doors, hatchways, etc., been closed prior to the collision, the *Victoria* would have continued to retain ample buoyancy and stability, and would not have ceased to be under control.

2. Even when so seriously injured and brought to such a critical condition, as was the case, had the turret ports and upper deck battery been closed, the armour door secured, and water excluded from the turret and battery, the *Victoria would not have capsized.* It is possible that she may have eventually foundered in consequence of the gradual passage of water into the forward compartments.

3. That under the serious circumstances of this collision, or of any similar accident which may occur, the safety of a ship and her continued flotation, demand that provision should be made for closing gun-ports and openings in upper works, through which water may pass into the interior of the ship, if the flooding of

the compartments produces great change of trim or serious heeling.

If such precautions are not taken, the virtual height of freeboard is reduced to the height of sills or doors, and the presence of the superstructures, when water is not excluded from them, does not assist either buoyancy or stability to any sensible extent.

For a detailed account of the above, see—

(i.) "Ironclads in Action." H. W. Wilson.

(ii.) "Life of Admiral Tryon." Admiral Fitzgerald.

(iii.) *Brassey's Naval Annual*, 1894.

(iv.) Parliamentary Paper, No. C. 7208, of 1893.

(v.) *Engineer*, November 10, 1893.

APPENDIX

QUESTIONS.

Chapter I.

1. Distinguish between the terms "structural" and "local" strains as applied to a ship. Enumerate a number of "local" strains. Have any of these local strains been sufficiently great in your experience to cause damage?

2. State the reasons for the superior efficiency of a I beam of steel to a solid rectangular beam of wood.

3. How may a ship be compared to a beam, and what parts of the structure are most efficient from this point of view?

4. How are the longitudinal strains on a ship's structure made the subject of calculation?

5. Why is the structure at the keel and at the upper deck considerably stronger in a long cruiser than in a battle-ship of the same total displacement?

6. Why is it that the boat deck and the topside plating adjacent are not made an integral part of the structure in a ship having a boat deck?

7. To what special sort of strain are the flat portions of a ship forward specially liable? Why do you consider that this straining action is less in evidence in war-ships than in merchant steamers?

8. Why is it possible to build a steel or iron ship considerably lighter than a ship of the same size built of wood?

9. Why must special attention be devoted to the strength of the upper deck and structure adjacent in a vessel of large proportion of length to depth? From this point of view, show that the most recent method of protection of large cruisers is more likely to prove economical as regards weight of hull structure than that adopted in, say, the cruisers of the *Edgar* class.

10. Suppose one had a vessel 300 ft. long, the structure of which had proved sufficiently strong, and a vessel of the *same depth*, but 360 ft. long were required. Discuss generally what portions of the structure would have to be strengthened to ensure the new vessel being sufficiently strong.

11. Indicate how the inspection and maintenance of a ship influences the design of the structure.

Chapter II.

1. State the qualities of "mild steel" that make it a suitable material for shipbuilding purposes.

2. Compare in tabular form the tests laid down for "mild steel," "rivet steel," "cast steel."

3. What tests are necessary in a steel casting beside those relating to the strength and ductility of the material? Why are such tests of great importance for castings of steel?

4. Compare the tests for "mild steel" and those for the special steel used in cruisers and destroyers.

5. It is laid down that holes in high-tensile steel must be *drilled*, and not punched. Why is this?

6. Describe the process of "pickling" steel plates. What trouble would you expect to arise in a ship's structure from the steel of which the "mill scale" had not been removed before painting?

7. Draw out to a large scale the section of a zed bar, a tee bulb, an angle bulb, and a I bar. State places in your present ship in which these sections are used.

8. A flange is frequently used on the edge of a plate instead of an angle bar for connection purposes. What advantage is thereby secured? State places in your present ship where this is done.

9. Describe the most ordinary form of rivet used in ship work, and show how such a rivet is used for the outer bottom plating where the outside surface must be flush.

10. What is meant by the *pitch* of rivets? State the amount of this pitch for rivets $\frac{7}{8}$-in. diameter where the work has to be watertight. What pitch would be used for $\frac{3}{4}$-in. rivets for internal work not watertight?

Ans. 4 to $4\frac{1}{2}$ in.; $5\frac{1}{4}$ to 6 in.

11. When your ship is next in dry dock, examine the "lap" caulking and the "butt" caulking of the outer bottom plating.

12. State the various advantages that result from ordering plating by the weight required per square foot rather than by thickness.

13. Taking the area of the outer bottom plating of a vessel as 30,000 square ft., estimate the saving of weight, if the steel plating is ordered 20 lbs. per square ft. instead of $\frac{1}{2}$ in. thick. What further saving would be possible if the manufacturer sends in all the plating down to the limit allowed, viz. 5 per cent. under? *Ans.* 5·4 tons; 13·4 tons.

14. If the area of the outer bottom plating (specified of 15 lbs.) is 20,000 square ft., what variation of weight is possible, in view of the latitude allowed to the manufacturer? *Ans.* About $13\frac{1}{2}$ tons.

15. What is annealing? What is the effect of annealing on a plate which has had a large number of holes *punched* in it?

Chapter III.

1. Distinguish between "bracket frame" and "solid plate frame." Where are these frames used in a large armoured ship?

2. Describe generally the construction adopted in battle-ships below armour within the limits of the double bottom.

3. What is a "floor-plate"? Sketch and describe the framing of a battle-ship before and abaft the double bottom.

4. State the advantages of having a double bottom to a ship. Why is it not possible to provide a double bottom in the smaller ships of the Royal Navy? Discuss the question of fitting a wing bulkhead below the armour deck in a large cruiser.

5. Why is the "transverse" system of construction more suited to the ends of a large ship than the "longitudinal" system?

6. Draw out a table with rough sketches showing the supporting frames for 4-in., 6-in., 9-in. armour. Why is a rigid support of great importance behind armour?

7. Draw in outline midship section, describe and compare the main features of (1) Admiral class, (2) *Royal Sovereign*, (3) *Majestic*, (4) *King Edward VII.*

8. Draw in outline midship section, describe and compare the main features of the first class cruisers (1) *Edgar*, (2) *Diadem*, (3) *Cressy*, (4) *Monmouth.*

9. Draw in outline midship section, and compare the main features of two second class cruisers. Show how the intended service has had a distinct influence on the design.

10. Draw in outline the midship section of a "sloop," and point out what provision is made in such a vessel for protective purposes.

11. State in general terms the distinction between a battle-ship and a first class cruiser. Compare H.M.S. *Triumph* with H.M.S. *Duncan* and H.M.S. *Cressy*, and state in what category you consider she should be placed.

12. Discuss the question of working zed bars for the framing of ships instead of two angles riveted back to back from the point of view of (i.) economy of weight, (ii.) saving of cost.

13. Discuss the importance of avoiding discontinuity of strength in a ship's structure.

14. Name typical vessels of the Royal Navy which are *sheathed* with wood and copper. Why have all the sloops and most of the second class cruisers built in recent years been sheathed?

15. In going through the double bottom of a vessel it will be noticed that the non-watertight longitudinals have no lightening holes in certain frame spaces. Trace the reasons for this.

16. Taking the length of the double bottom of a battle-ship as 250 ft., and the frame spacing 4 ft., make an estimate of the saving of weight if an oval manhole 23 in. × 15 in. is cut in *every* frame space in the non-watertight longitudinals. *Ans.* $5\frac{1}{2}$ to 6 tons.

17. What advantages beside saving of weight are obtained by cutting lightening holes in a longitudinal girder?

18. State places in your present ship where you have noticed holes cut for lightening purposes. Do you consider any weakness has resulted from the removal of this material?

19. What is a *middle-line kelson?* When this is intercostal, and the floor

plate is continuous from side to side, how is proper continuity of longitudinal strength obtained?

CHAPTER IV.

1. Draw out the sections of the beams used in your present ship for the various decks and platforms. State the spacing of the beams amidships and at the ends.

2. Under what circumstances are zed bars likely to be advantageous for beams? Why do you consider it would have been inadvisable to form the beams to the decks of your ship of zed bars?

3. Why is the connection of beams to frames of great importance? Draw out two such connections as fitted in your present ship.

4. What is a carling? State places in your ship where carlings are fitted.

5. Why should the upper deck of a vessel be given a *round up* while the platform deck is worked level?

6. Why would a log of timber wedged up between-decks be inefficient as a pillar from a structural point of view?

7. Name places in your ship where the pillars have had to be made "portable."

8. In sailing ships of the mercantile marine in which only one bulkhead (the collision bulkhead) is fitted, it is laid down in "Lloyd's Rules" that the beam arms have to be three times the depth of the beam. Discuss the reasons for this, bearing in mind the ordinary practice in ships of the Royal Navy.

9. If the upper deck of your ship is completely plated, trace the bolt fastenings to the planks of the wood deck. See also where these bolts are placed to take the butts. If the deck is not completely plated, see how the butts of the planks are fastened.

10. Why are pillars always made hollow in vessels of the Royal Navy?

11. Notice in your ship the arrangement of butts of "sheer strake," "stringer plate," and the structure adjacent. See how a good shift of butts is obtained. Why is this of great importance?

12. What do you consider the advantages and disadvantages of *corticine* as a covering for decks in place of wood? How is this corticine secured?

13. Describe the operation of "caulking" as applied (*a*) to steel plating, (*b*) to a wood deck. In both cases state what preparations are necessary before the caulking is started. What essential quality is caulking intended to secure?

CHAPTER V.

1. What special advantages attach to the use of steel instead of iron for the skin plating of a vessel?

2. Trace in Fig. 46 how the strength of the structure is made as uniform as possible, no special frame space being a special place of weakness, but a good shift of the butts being obtained.

3. What is the "raised and sunken" system of working the outer bottom plating? What is the function of the "liner"?

4. Sketch a "lightened liner." Does the lightening in any way injure the strength of the structure?

5. State places where the outer bottom plating of a large ship is *doubled*. For what purpose is this done in each case?

6. Why is the transverse section of a ship in way of a watertight bulkhead a line of weakness? Show how this weakness is compensated for.

7. Sketch the form of *bulkhead liner* adopted in your present ship.

8. State the requirements necessary in a watertight manhole to a double-bottom compartment. Draw out to a large scale the plug fitted in the cover, and state its uses.

9. Make a tabular statement giving the thicknesses of keel, outer bottom plating, and sheer strake in (*a*) a battle-ship, (*b*) a second class cruiser, (*c*) a third class cruiser, (*d*) a sloop, (*e*) a destroyer.

10. What is a "liner"? What purpose does it serve? Would there be any objections to making such a liner of iron instead of steel if desired on the score of cheapness?

Chapter VI.

1. What is a "collision" bulkhead? Why is this bulkhead of great importance?

2. How far is the collision bulkhead from the forward perpendicular in your ship? How does this compare with the distance required for merchant ships, built to the rules of "Lloyd's Register," viz. one-twentieth the length?

3. Why do you think the rules of the Registration Societies specify the minimum distance of the collision bulkhead abaft the stem?

4. State the special advantages attendant on the provision of a watertight bulkhead between the engine-rooms.

5. For what reason is the stiffening adopted for the main bulkheads of a large ship of so strong a character? Why can the bulkheads at the ends of the machinery space be safely built with considerably less stiffening than those between the boiler-rooms?

6. How are compartments of a ship tested for watertightness while building? What does this testing ensure besides watertightness?

7. Write out the instructions regarding the tests required for watertight compartments of ships in commission. (See Admiralty Circular, S. 32111/1903, January 29, 1904.)

8. What is a "cofferdam" bulkhead? Is one fitted in your present ship? If so, state the means of access to the space between it and the collision bulkhead.

9. When a magazine is placed next to a boiler-room it is usual to provide an air space between to prevent the magazine becoming heated. Is this the case in your present ship? If so, how is access obtained to the space? Why is such access necessary?

10. Trace the means of escape under protection fitted in your ship from the submerged torpedo-rooms. Why is such escape necessary?

11. Make a list of the horizontal sliding watertight doors in your ship. In each case see if it would have been practicable to fit a vertical sliding door instead.

12. Make a list of the most important vertical sliding doors in your ship. In each case state why a hinged door would have been undesirable.

13. How many double-bottom compartments (excluding the wings and reserve feed spaces) are there in your present ship? What is the water capacity in tons of the largest, and what of the smallest? What is the total capacity in tons of this double bottom available for the introduction of water as ballast?

14. Investigate the means of access in your present ship to—

(i.) All wing compartments;
(ii.) Spaces below provision rooms, etc., forward and aft;
(iii.) Spaces before the collision bulkhead;
(iv.) Inside of masts;
(v.) Watertight compartments at the sides of submerged torpedo-room.

Why are such means of access provided to such spaces not used for stowage purposes?

15. Describe with outline sketches the three types of watertight door fitted in ships of the Royal Navy. What are the conditions governing the adoption of each type?

16. Why is it essential that when name plates or fittings are removed from bulkheads that the holes left should be immediately filled up by tap rivets? (See Admiralty Circular, S. 32111/1903, January 29, 1904.)

Chapter VII.

1. Sketch and describe the stem and its supports for a battle-ship. Draw special attention to the provision that is made to withstand the side bending that is likely to take place when the ships swing together after ramming.

2. Sketch and describe the stem for a *sheathed* vessel.

3. What are the special functions of the sternpost of a twin-screw vessel? Sketch and describe such a sternpost with the connection of the adjacent structure.

4. What special features do the sternposts of modern cruisers possess? Make a sketch of one such sternpost. What is specially neccessary for the support of the rudder?

5. What information is supplied to your ship with respect to the rudder?

6. Make a sketch of a shaft bracket. Indicate the exact nature of the strains that this fitting has to withstand. What is it made of in your ship?

7. Make a sketch of rudder of either—

(i.) A battle-ship; or,
(ii.) A cruiser.

8. Why is it necessary to make the diameter of the rudder of a cruiser of considerable size, in view of the fact that the twisting moment is small, even at high speeds, because the rudder is balanced?

9. Compare the pressure *per unit area* on a rudder in—

(i.) A 10-knot cargo steamer;
(ii.) A 19-knot battle-ship;
(iii.) A 23-knot cruiser;
(iv.) A 30-knot destroyer.

Ans. 1 : 3·6 : 5·3 : 9·0.

10. Indicate in detail how the weight of a rudder is taken in (i.) a battle-ship like *Duncan* and (ii.) a cruiser with a balanced rudder.

11. State in detail what would need to be done to remove the rudder of your ship. How can it be lifted and withdrawn for examination and repair?

12. Show how the hole in the sternpost for the reception of the rudder head is made watertight.

13. Sketch the outline of the sternpost and rudder in *Edgar*, *Cressy*, *Duncan*, and *King Edward VII.*

14. Name any ships you know of in the Royal Navy which have two rudders. Sketch the shape of the stern of one of these ships.

15. State the essential conditions to be satisfied in designing the stern of a large war-ship.

CHAPTER VIII.

1. State what is meant by a "compensating" steering gear. How is compensation obtained in "Rapson's slide," "Harfield's gear," and "Ollis's gear"?

2. What special advantages attach to the use of the screw steering gear?

3. What advantage is there in a compensating steering gear in view of the desirability of being able to steer the ship by manual power?

4. In Fig. 86 of Harfield's gear it will be noticed that the diameter of the spindle to the forward cross-head is considerably less than the diameter of the rudder-head. Why are these diameters so different, seeing that both have to take the same twisting moment?

5. In Harfield's and Rapson's slide steering gears (Figs. 86 and 84), why are not the tillers keyed direct on to the rudder-head without the use of the connecting rods?

6. Is a "Fayrer's" brake fitted to the hand-wheels of your ship? If so, what purpose does it serve? If not, why has it been dispensed with?

7. Why is it essential that steering gear should be under perfect control when changing from steam to hand, or from one steam engine to the other?

8. What alternative methods of steering your ship are possible, supposing the steering gear aft and the rudder are in order?

9. Supposing the rudder of your ship carried away, how could you steer the ship?

10. Make a list of the positions from which your ship can be steered, supposing the steam steering engines are available.

Chapter IX.

1. Make a list (with capacities, if possible) of all the pumps available in your ship for dealing with a leak.

2. Show that the initial rate of inflow of water through a hole d ft. below the surface is about $14\sqrt{d}$ tons per hour for every square foot of area of the hole.

3. Explain how the centrifugal circulating pumps in the engine-rooms may be made available for dealing with inflow of water into a ship.

4. Sketch the "main drain" as fitted to a modern ship. Show how water is prevented from passing away from the engine-room or from one boiler-room to another through this main drain.

5. How is water got rid of from (i.) wings, (ii.) barbettes, (iii.) submerged torpedo-rooms, (iv.) chain lockers?

6. Sketch and describe a short portion of the "main suction." What special type of valve is necessary for the suction from a double-bottom compartment, and why?

7. Suppose it has been necessary to flood a magazine in your ship. How would the water be got rid of?

8. Sketch in outline and describe the construction and working of a Downton pump.

9. Sketch and describe a Kingston valve. How many of these valves are fitted to your ship? Make a list stating the purpose of each of them.

10. How is water collecting on the top of the inner bottom got rid of?

11. State exactly what you would do to pump out water in, say, the forward boiler-room, supposing steam is not available.

12. Trace what it would be necessary to do to flood the largest magazine forward in your ship.

13. Why are valves for opening a magazine to the sea kept locked? Where are the keys kept?

14. Sketch an air escape as fitted to (i.) a magazine, (ii.) a wing compartment.

15. Sketch a portion of the "fire-main," with a specimen "rising main." What is the fire-main used for?

16. How would you get water for wash-deck purposes or for fire, in the event of the steam pumps not being available?

17. Sketch and describe the fittings in your ship for flooding magazines, etc., when the ship is in dry dock.

18. The "main drain" was formerly led through the double bottom. What advantages result from the present practice of leading the main drain *above* the inner bottom?

Chapter X.

1. Compare the ventilation for an ordinary building and a ship.

2. Why is continuous and efficient ventilation of coal-bunkers of great importance? Write out the regulations regarding the ventilation, etc., of coal-bunkers contained in the "Steam Manual."

3. Sketch and describe the ventilation of (1) an upper bunker, (2) a lower bunker.

4. Describe generally the system of ventilation with large steam-driven fans. What disadvantages are connected with this system?

5. Describe generally the present system of ship ventilation.

6. Describe, with sketches, how one of the largest magazines forward in your ship is ventilated. Why are ventilation exhausts fitted to magazines and not to shell-rooms?

7. Write out the regulations contained in the "Gunnery Manual" regarding the ventilation of magazines.

8. Why is it undesirable to paint the ends of magazine exhausts and air-escape pipes?

9. Describe, with sketches, the ventilation in your ship of (i.) sick bay, (ii.) spirit room, (iii.) engineer's workshop, (iv.) capstan engine department.

Chapter XI.

1. What is rust, and under what conditions is it formed?

2. Describe the process of "pickling." What trouble would you expect to arise in a ship from the steel of which the mill scale had not been removed before painting?

3. Why is it essential to keep the steel of a ship always well painted?

4. What is a zinc protector? What object does it serve? Why must zinc protectors never be painted?

5. State places in your present ship where you have observed zinc protectors fitted.

6. What is fouling? How is fouling prevented in (i.) a wood ship, (ii.) a steel ship?

7. Sketch and describe the present system of sheathing vessels with wood and copper. What service are such vessels specially suitable for? Name vessels of the Royal Navy thus sheathed.

8. What is *exfoliation*?

9. For what purpose is cement fitted in ships? Under what circumstances would cement prove detrimental rather than beneficial?

10. What is *cork cementing*? What is the object of this process?

11. State the regulations laid down in the "Steam Manual" respecting the inspection of the structure of H.M. ships.

Chapter XII.

1. Explain generally the action of a Temperley transporter.

2. Take one outer upper bunker of a ship and show (1) how coal is got into the bunker, (2) how the coal is got into the lower bunkers from it.

3. Take 30 ft. of the main deck of your ship and make a drawing showing all the coaling scuttles and escapes that are fitted, and state the purpose of each.

4. Sketch and describe a *screen* to a coal-bunker door. Why are such screens fitted?

5. What are temperature tubes, and why are they fitted? Write out the regulations in the "Steam Manual" respecting their use."

CHAPTER XIII.

1. Describe with outline sections the armour or other protection in (1) a battle-ship, (2) a deck-protected cruiser, (3) a sloop.

2. Discuss briefly the circumstances which led up to the particular arrangement of protection adopted in the *Inflexible*.

3. What is *compound* armour, *Harveyed* armour, and *Krupp* armour. Compare the relative efficiency of these types of armour.

4. Trace the reasons which led to the adoption of armour for the first class cruisers of the *Cressy* class.

5. Describe the system of protection adopted in (i.) *Royal Sovereign*, (ii.) *Majestic*. Compare and contrast the two systems.

6. For what purpose is teak backing fitted? What is the thickness now adopted?

7. Sketch an armour bolt for hard-faced armour. What is the object of the sleeve, the india-rubber washer, and the cup washer?

8. What is meant by the term "figure of merit" as applied to armour?

9. Describe the armour protection, etc., of a vessel of the *Duncan* class.

10. What is a cofferdam? State places in your ship where cofferdams are fitted.

11. What is the value of coal as a material for purposes of protection? What vessels depend wholly on their coal for protection?

12. In the *Duncan* the barbettes are 11 in. in thickness, while the side is only 7 in. State the reasons for this difference.

13. Compare the method of protecting the gun mountings for heavy guns in (1) *Admiral* class, (2) *Royal Sovereign*, (3) *Duncan*.

14. Make out a list of the armament in your present ship, stating in each case how the rear of the gun, and how the mounting, etc., is under protection.

15. What are "armour gratings"? Why are they fitted? State places in your present ship where you have noticed armour gratings fitted.

CHAPTER XIV.

1. A hollow pillar is 4 in. external diameter and $\frac{3}{8}$ in. thick. What is the sectional area, and what would be the weight in pounds of 10 ft. of this pillar, if of wrought iron (480 lbs. to cubic foot)?

Ans. 4·27 square in.; 142 lbs.

2. A wrought-iron armour plate is 15 ft. 3 in. long, 3 ft. 6 in. wide, and $4\frac{1}{2}$ in. thick. Calculate its weight in tons. *Ans.* 4·29 tons.

3. Steel armour plates (490 lbs. to the cubic foot) are demanded 400 lbs. per square foot instead of 10 in. thick. What is the saving of weight per 1000 square ft. of surface of this armour? *Ans.* 3·7 tons.

4. A mast, 90 ft. in length, and 3 ft. external diameter, is composed of 20-lb. plating, worked flush jointed (as Fig. 8) on 3 tee bars, 5 in. × 3 in. × $15\frac{1}{2}$ lbs. per foot. Estimate the weight. *Ans.* About $9\frac{1}{2}$ tons.

5. A curvilinear area has the following ordinates at equidistant intervals of 18 ft., viz. 6·2, 13·8, 21·9, 26·4, 22·35, 14·70, and 7·35 ft. Assuming

that Simpson's first rule is correct, find the percentage of error that would be involved by using the Trapezoidal rule. *Ans.* 1·2 per cent.

6. The semi-ordinates of the load water plane of a vessel are 0·2, 3·6, 7·4, 10·0, 11·0, 10·7, 9·3, 6·5, and 2·0 ft. respectively, and they are 15 ft. apart. What is the area of the load water plane? *Ans.* 1808 square ft.

7. The vertical sections of a vessel 10 ft. apart have the following areas: 10, 50, 60, 70, 50, 40, 20 square ft. Find the volume of displacement. *Ans.* 2966 cubic ft.

8. In a given ship, pillars in the hold can be either solid iron, $4\frac{3}{4}$ in. diameter, or hollow iron, 6 in. diameter, and $\frac{1}{2}$ in. thick. Estimate the saving of weight for every 100-ft. length of these pillars, if hollow pillars are adopted instead of solid, neglecting the effect of the solid heads and heels of the hollow pillars. *Ans.* 1·35 tons.

9. A bunker, 24 ft. long, has a mean section of the form of a trapezoid, with parallel sides 3 ft. and 4·8 ft., 10·5 ft. apart. Find the number of tons of coal contained in the bunker, at 43 cubic ft. to the ton.

Ans. 22·8 tons.

Chapter XV.

1. What is the displacement quoted in the "Navy List"?

2. Explain what is meant when a ship's dimensions are stated as follows in the Navy Estimates. Length, 300 ft.; breadth, 36 ft. 6 in.; draught forward, 12 ft.; aft, 15 ft.; displacement, 2,135 tons.

3. State the draughts of your ship in three distinct conditions of loading.

4. Define "deep load," "normal load," "light."

5. What is "gross register" and "nett register" as applied to the tonnage of war-ships? Add to the comparative table of tonnage in Chapter XV., the various tonnages for your present ship as given in the Ship's Book.

6. Compare the methods of referring to the tonnage of war-ships and merchant ships, and explain why it is that the great size of modern liners, as compared with the largest war-ships, is not appreciated because of the different systems of stating the tonnage.

7. Write out the regulations contained in the King's Regulations regarding the statements to be given of the tonnage of H.M. ships.

8. Why do ships of the Royal Navy have to be measured by the Board of Trade officers for tonnage?

Chapter XVI.

1. Explain and justify the statement, "the weight of a vessel is equal to the weight of the water displaced."

2. A vessel at Gravesend, where the water weighs 63·7 lbs. per cubic ft., is found to displace 60,500 cubic ft. What is her weight in tons?

Ans. 1720 tons.

3. What is the weight of a box-shaped vessel 150 ft. long, $17\frac{1}{2}$ ft. broad, which is found to float in salt water at a draught of 6 ft. 3 in.

Ans. $468\frac{3}{4}$ tons.

4. A cylinder is 500 ft. long, 20 ft. diameter, and floats with the axis in the water surface. Find its weight when floating thus in salt water.

Ans. 2244 tons.

5. State, if possible, the tons per inch of your present ship, and compare it with the approximation given in Chapter XVI.

6. Bilge keels are to be fitted to a ship whose tons per inch is 48. The estimated weight of the bilge keels is 36, and the volume they occupy is 840 cubic ft. What will be the increase of draught due to fitting these bilge keels? *Ans.* $\frac{1}{4}$ in.

7. What is the "coefficient of fineness?" Determine its value for the following ships:—

(i.) H.M.S. *King Edward VII.*, 425 ft. × 78 ft. × 26 ft. 9 in. × 16,350 tons.
(ii.) SS. *Umbria*, 500 ft. × 57 ft. × 22 ft. 6 in. × 9860 tons.
(iii.) H.M.S. *Duke of Edinburgh*, 480 ft. × 73$\frac{1}{2}$ ft. × 27 ft. × 13,550 tons.
(iv.) H.M.S. *Diadem*, 435 ft. × 69 ft. × 25 ft. 3 in. × 11,000 tons (keel projection 8 in.). *Ans.* (i.) 0·645; (ii.) 0·538; (iii.) 0·5; (iv.) 0·52.

8. What is "tons per inch immersion"? Find the value of this for a box-shaped vessel 150 ft. long, 30 ft. broad. *Ans.* 10·7 tons.

9. Draw out a curve giving the displacement of the vessel in the previous question for all draughts up to 12 ft.

Ans. Curve is a straight line from 0 at zero draught to 1543 tons at 12 ft. draught.

10. A vessel of 14 ft. mean draught has the following displacements at level lines 2 ft. apart, viz. 2118, 1682, 1270, 890, 553, 272, 71. Draw out the curve of displacement and state (i.) the displacement at a draught of 12 ft. 3$\frac{1}{2}$ in. forward, 13 ft. 6$\frac{1}{2}$ in. aft.; (ii.) the mean draught when the ship displaces 1750 tons; (iii.) the tons per inch immersion at 13 ft. draught.

Ans. (i.) 1880 tons; (ii.) 12 ft. 4 in.; (iii.) 18·1 tons.

11. The tons per inch of a vessel at waterlines 2 ft. apart are 19·45, 18·51, 17·25, 15·6, 13·55, 10·87, and 6·52, the lowest waterline being 18 in. above the underside of flat keel. Draw the curve of tons per inch immersion to scale and estimate the number of tons necessary to sink the vessel from a draught of 12 ft. to a draught of 13 ft. 6 in. *Ans.* 344 tons.

12. The area of a waterplane at which a ship floats is 6300 square ft. What will the sinkage be if 45 tons be placed on board? *Ans.* 3 in.

CHAPTER XVII.

1. Define "centre of gravity," "centre of buoyancy."

2. Write down the three necessary conditions that must be fulfilled for a ship to float freely and at rest in *stable equilibrium.*

3. Define stable and unstable equilibrium.

4. What is the *transverse metacentre?* How does its position with reference to the C.G. determine the stability of a ship for small angles?

5. Show how the position of the metacentre with reference to the C.B. depends on (i.) the shape of the waterplane and (ii.) the displacement.

6. What is the *metacentric height?* State this length for your ship in the deeply laden condition.

7. What is a *metacentric diagram?* Of what use is such a diagram when made?

8. Indicate the reason that a log of timber half immersed will not float with one face horizontal.

9. What is the inclining experiment? What knowledge of the ship does one obtain from this experiment?

10. Show that if a deflection a is observed in a length l of a pendulum, after traversing w tons through d ft. across the deck of a vessel W tons displacement, the metacentric height is given by—

$$GM = \frac{w \times d \times l}{W \times a}$$

11. A vessel of 1722 tons displacement is inclined by shifting 6 tons of ballast across the deck through $22\frac{1}{2}$ ft. A mean deviation of $10\frac{1}{2}$ in. is obtained with pendulums 15 ft. long. The transverse metacentre is 15·28 ft. above keel. Show that the C.G. of the ship is 13·95 ft. above keel.

12. Discuss the importance of coal-bunkers above the protective deck of a deck-protected cruiser from the point of view of stability.

13. What effect has *girdling* at the waterline on the stability of a ship?

14. Explain how it is that a ship when partially waterborne, as when being dry docked, suffers a reduction of stability. Under what particular combination of circumstances may this reduction be sufficient to cause instability?

15. A vessel of 1792 tons displacement is inclined by shifting 5 tons already on board transversely across the deck through 20 ft. The end of a plumb-line 15 ft. long moves through $5\frac{1}{4}$ in. Determine the metacentric height. *Ans.* 1·91 ft.

16. A two-masted cruiser of 5000 tons displacement has its C.G. 2 ft. above the waterline. It is decided to add a military top to each mast. Assuming the weight of each top with its guns, men, and ready ammunition supply to be 12 tons, with its C.G. 70 ft. above the waterline, what will be the effect of the change on the metacentric height?

Ans. Reduce about 0·3 ft.

CHAPTER XVIII.

1. Define *longitudinal metacentre.* Write down and explain the formula for obtaining its position with reference to the C.B.

2. Define *trim, change of trim.* Write down an approximate formula, suitable to your ship, for the moment to change trim 1 in.

3. Make an estimate of the change of trim in a ship of *Pelorus* class due to moving 10 tons through a fore-and-aft distance of 200 ft.

Ans. About 7 in.

4. State the trim of your ship in the *deep load* condition. What is the trim in the most usual seagoing condition?

5. Are any trimming tanks fitted in your ship? If so, explain under what circumstances you would fill them.

6. Show why it is that many ships floating on an even keel will go down by the head if a weight is placed at the middle of the length.

7. Explain how it is possible to determine the position of coals on board a ship such that the draught aft will not vary whether they are in or out.

8. Find the moment to change trim 1 in. of a vessel 400 ft. long, having given the following particulars: Longitudinal metacentre above C.B., 446 ft.; distance between C.G. and C.B., 14 ft.; displacement, 15,000 tons.

Ans. 1350 foot-tons.

9. A log of fir, specify gravity 0·5, is 12 ft. long, and the section is a square of 2 ft. side. Find the longitudinal metacentric height when floating in *stable equilibrium*. *Ans.* 16·5 ft. nearly.

10. H.M.S. *Hermes* is 350 ft. × 54 ft. × 5600 tons when floating at a draught of 19 ft. 6 in. forward, 21 ft. 6 in. aft. It is desired to take her over a bar at which the depth of water is 19 ft. Approximate to the weight necessary to remove, and where the C.G. of this weight should be.

Ans. 570 tons removed from the ship so disposed that the C.G. is about 11 ft. forward of amidships.

11. A vessel whose "moment to change trim 1 in." is 73 foot-tons, floats at a draught of 6 ft. 6 in. on an even keel. Determine the draught forward and aft if a weight of 5 tons is moved aft through a distance of 135 ft.

Ans. 6 ft. $1\frac{1}{4}$ in. forward; 6 ft. $10\frac{3}{4}$ in. aft.

12. A battle-ship 400 ft. × 75 ft. floats at a draught of 26 ft. 3 in. forward, 27 ft. 3 in. aft, and displaces 15,000 tons. Make an estimate of the new draught if 500 tons of coal is added 75 ft. before amidships.

Ans. Forward, 28 ft. $6\frac{1}{2}$ in.; aft, 26 ft. $8\frac{1}{2}$ in., about.

13. At about what station of your ship would the effect of added weight be to increase draught bodily but not to change the trim?

14. What relation exists between the transverse and longitudinal stability of a wholly submerged body? Discuss the question of submarine navigation from the point of view of longitudinal stability.

Chapter XIX.

1. What are the important features of a curve of stability?

2. Why is it specially desirable to have the curve of stability of a sailing-vessel enclosing a large area?

3. What is the angle of maximum stability of your ship in the "deep load" and "normal load" conditions? What is the "range of stability" in both of these conditions?

4. A vessel of 5000 tons displacement has a metacentric height of 2·0 ft. What effort does she make to return to the upright when inclined to (i.) 3°, (ii.) 6°, (iii.) 9°? *Ans.* 523; 1045; 1564 foot-tons.

5. A vessel's curve of stability has the following ordinates at angles of 15°, 30°, 45°, 60°, 75°, viz. 0·51, 0·97, 0·90, 0·53, 0.08 ft. respectively. Estimate the influence on the range of stability caused by lifting the C.G. of ship 0·2 ft. *Ans.* Reduce nearly 6°.

6. Draw out curves of stability of two ships, one having a great metacentric height and one having a moderate metacentric height.

7. What information is supplied to your ship as to the conditions of stability in various conditions?

8. What is the effect of excessive "tumble home" on the stability of a ship at large angles?

9. State the various assumptions which have to be made when making the stability of a ship at large angles the subject of calculation. Do you think any of these assumptions are unreasonable?

10. In the refitting of the *Royal Sovereign* class with casemated guns on the upper deck, it has been found undesirable to make any alteration to the turret ship *Hood*. What reason do you think has led to this action in this special ship of the class?

CHAPTER XX.

1. Quote a formula giving the period of oscillation of a ship when rolling unresistedly in still water.

2. What is "isochronous" rolling?

3. What is the "period" of your present ship?

4. Why was it that the *Inflexible* had the short period of 5¼ seconds? Why is such a short period undesirable?

5. Explain generally the action of bilge keels in reducing rolling.

6. Sketch the standard form of bilge keel as fitted to large steel ships of the Royal Navy.

7. Why is it that a ship of large metacentric height is likely to be a bad sea-boat, and a ship with small metacentric height a good sea-boat?

8. In a ship rolling among waves, what is meant by the "virtual upright" at any particular instant?

9. A ship of 6 seconds' period is placed broadside on to a regular series of waves having a period of 12 seconds. What will happen?

10. Justify the statement that a "*crank* ship is likely to be exceedingly steady in a sea-way."

11. What is *synchronism*? If the ship in question 9 is under control, what could the commanding officer do to destroy the synchronism?

12. Explain the principle of Mallock's rolling indicator.

13. Why is a pendulum untrustworthy in regard to giving indications of the rolling of a ship?

CHAPTER XXI.

1. Compare the influence of a "balanced" rudder on the turning of a ship as compared with an "unbalanced" rudder of the same area.

2. In some ships it is noticed, that on first putting the rudder over, the ship bends away from the straight before taking the spiral course. Give an explanation of this.

3. Explain the features of the Japanese battleship, *Yashima*, which caused her to have an outward heel of 8¾° when on the circle at full speed. Have you noticed any similar phenomenon in your experience?

4. What is the "pivoting point" of a ship when turning? Can you locate this point in your present ship?

5. What difference would you expect to find in your present ship if turning at full speed (*a*) with steam steering engine in use, (*b*) with hand-wheels only available? Also what difference at 10 knots under these circumstances?

6. Show that the influence of the after deadwood of a ship is favourable to turning when the rudder is first put over, but unfavourable when on the circle.

7. Enumerate all the advantages you think a twin-screw ship possesses in comparison with a single-screw ship.

8. State for your present ship how an object ahead could be best avoided, either by—

(*a*) Turning with full helm, both screws ahead;
(*b*) Turning with full helm and one screw astern; or
(*c*) Reversing both engines.

CHAPTER XXII.

1. What is *effective horse-power?* The *Greyhound* was towed at the rate of 845 feet per minute, and the horizontal strain on the tow-rope, including an estimate of the air resistance of masts and rigging, was 6200 lbs. Find the E.H.P. at that speed. *Ans.* 159 H.P. nearly.

2. Suppose we took a destroyer of 250 tons displacement and 27 knots speed as a model, and designed a vessel of 10,000 tons displacement of *similar form*. At what speed of this vessel could we compare her resistance with that of the model at 27 knots? (The ratio of length will be $\sqrt[3]{\frac{10000}{250}}$)

Ans. 50 knots.

3. A vessel of 7000 tons requires 10,000 I.H.P. to drive her 20 knots, and the I.H.P. at that speed is varying at the fourth power of the speed. Find approximately the I.H.P. necessary to drive a similar vessel of 10,000 tons at a speed of $21\frac{1}{2}$ knots. *Ans.* 16,000 I.H.P.

4. Make out a table for your present ship, giving speeds and corresponding horse-power. Construct a curve of power on base of speed, as Fig. 205, and make out a table, as for *Drake*, in Chapter XXII., giving increments of power for every knot from 10 knots to the highest speed.

5. In the previous question, do any of the spots come manifestly below the general run of the curve? Are any of these due to (*a*) foulness of bottom, (*b*) bad weather, or (*c*) running in shallow water?

6. Enumerate the various factors which make up the difference between the E.H.P. and the I.H.P. of a ship.

7. Explain the reasons why it is of great importance to keep the bottoms of war-ships clean. How often is your present ship put into dry dock?

8. If it were possible to instal enough power to drive a submarine at high speed, would you not expect that the speed below water would be greater than the speed on the surface, in view of the absence of wave-making resistance?

9. Which is the more economically propelled at full speed, a duck or a fish?

10. Suppose one engine of *Drake* broke down, at what speed do you

estimate she could go, (*a*) "with all despatch," (*b*) "with despatch," (*c*) "with moderate despatch?"

Ans. (*a*) $18\frac{1}{2}$–19 knots; (*b*) 17–$17\frac{1}{2}$ knots; (*c*) $14\frac{1}{2}$–15 knots.

11. A twin-screw Atlantic liner in mid Atlantic breaks one propeller shaft, and it is stated that she finishes the voyage with one engine at three-quarter speed. How do you account for a falling-off of only *a quarter* the speed, with a reduction of *one-half* the power?

CHAPTER XXIII.

1. Discuss in general terms the difference between the conditions under which a naval architect has to design and build a war-ship and an architect to design and build a large building, or a civil engineer a bridge.

2. Discuss in general terms the conditions under which a war-ship designer has to work as compared with a naval architect designing large steamers of the Mercantile Marine.

3. State the advantages of having a forecastle in a war-vessel designed for high speed.

4. What advantages from a gunnery point of view are found in a war-ship of high freeboard, in addition to the advantages of comfort and seaworthiness?

5. Make a comparison between the dimensions and particulars of H.M.S. *Drake* and *Duncan* of nearly the same displacement, and indicate as far as you can the reasons for the differences in each case.

6. State the reasons which have caused large Atlantic liners to be considerably larger than the largest cruisers.

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THE END.

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